MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

Ruling mechanism on Gaertner Scientific Corp. circular dividing machine. Indexing head may be set to rule accurate lines to any graduated units desired. Respective pins drop into head to control length of graduated lines.

CONTENTS

January, 1948   Vol. 20, Number 1

Itemized Index

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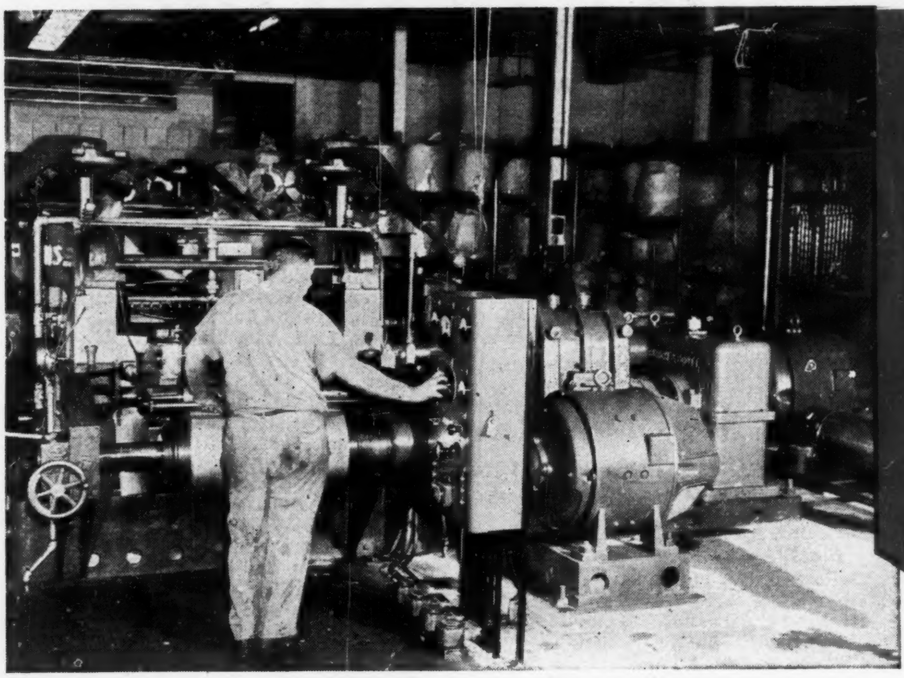

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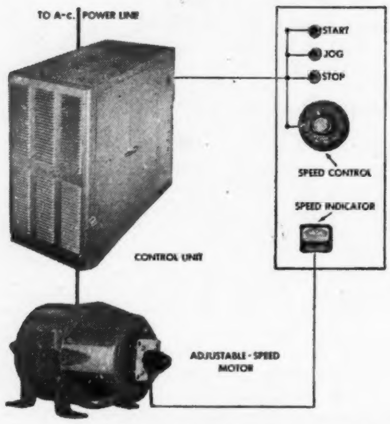

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RELIANCE A-C D-C MOTORS

"Motor-Drive is More Than Power"

Topics

GLASS CLOTH laminates have been made transparent, according to the General Electric Co. The polymer used in bonding is a new material, diallyl phenyl phosphonate. It wets the glass fibers sufficiently and its index of refraction so nearly matches that of glass as to give the laminate transparent properties.

AUTOMATIC FOCUS RECTIFIER for mosaic map making has been built for the Army Air Corps by Bausch & Lomb. It simultaneously enlarges and prints aerial photographs automatically transforming them to a common scale and level. This corrects the tilt encountered as a result of the plane's variation in angle and level at the time photographs are taken.

TUBELESS TIRES for automobiles combine the features of puncture-sealing tubes, improved riding qualities, high bruise resistance, and ability to retain air pressure. Embodying rayon cord construction, the tires developed by The B. F. Goodrich Co. are in test service on a taxicab fleet, state police cars and a number of passenger cars.

HIGH-TEMPERATURE FURNACE capable of producing temperatures of 3100 F in a zone of about four cubic feet has been built by Westinghouse for high-temperature research. It is a radiant type furnace using molybdenum rods as heating elements, laid on ledges in the refractory brick. In another furnace temperatures approaching 5000 F will be sought. In it a tungsten crucible, large enough to hold a walnut, is heated in an inert gas by combination of radiant heat and high-frequency induction.

ELECTROSTATIC ACCELERATOR, a type of atom smasher, is being constructed by General Electric Co. for the Brookhaven National Laboratory at Upton, N. Y. One of the several electronuclear machines planned for the laboratory, the machine is rated at 3,500,000 electron volts and will be used for "source focusing" studies to determine, in part, the designs of other larger machines and associated equipment for the laboratory. Later it will be used as a source of high-energy particles to be directed in a beam at target elements for fundamental studies.

SYSTEM FOR CALIBRATING the diaphragm openings of a photographic lens, developed at the National Bureau of Standards, takes into account the losses of light from absorption, reflection and scattering within the lens instead of being a mere geometric ratio between the lens diameter and its focal length as commonly used.

BROACHING of narrow openings in stacks of stampings, such as cores of small motors, is considered by the General Electric Co. as the most effective method of removing burrs and small irregularities from this type of assembly.

GYRO-GLIDER, an odd-looking wingless aircraft, works like an autogiro without a power source. Weighing 120 pounds or 1/3 the weight of a standard glider, it is flown with the aid of two 9-foot rotating blades and is capable of lifting a load of nearly 300 pounds.

RUSTPROOFING treatment of steel using a phosphate coating process developed by Calgon Inc. resists salt-fog tests up to 1000 hours and withstands, without blistering or corrosion, 1000-hour exposure in 100 per cent humidity at 100 F.

SOUTHWEST Research Institute, formed to aid the section's industry solve its technological problems, has an additional objective of furthering friendly relations with Latin-American countries by bringing in science students from those countries to work in the institute.

MACHINE DESIGN

Improved Metals Promise DESIGN PROGRESS

BECAUSE of new machine developments and also because of the beginning of new industries that have created new and special requirements for materials of construction, some relatively new metals and metals applications are of special interest to the machine designer. To enable the designer to evaluate and determine whether these new materials can be used to advantage in his machines, an attempt will be made to cover the various improvements and developments that have taken place during the past few years in the field of iron, nickel and cobalt-base metals as well as some of the other special materials.

***NEW AND IMPROVED METALS** are of primary importance in the development of machines capable of operating successfully under today's exacting requirements. It is believed that the items covered in this article represent the most important of these innovations, the aim being to present information such as will enable the designer to intelligently seek out those metals most suitable for specific machine applications*

LOW-ALLOY, HIGH-STRENGTH STEELS: Development of light-weight, high-strength structures by the aircraft industry, *Fig.* 1, has brought about a great change in view-

Fig. 1 — Below — Light-weight, high-strength aircraft structures have created a considerably changed viewpoint regarding the use of metals in machines. Photo, courtesy Glenn L. Martin Co.

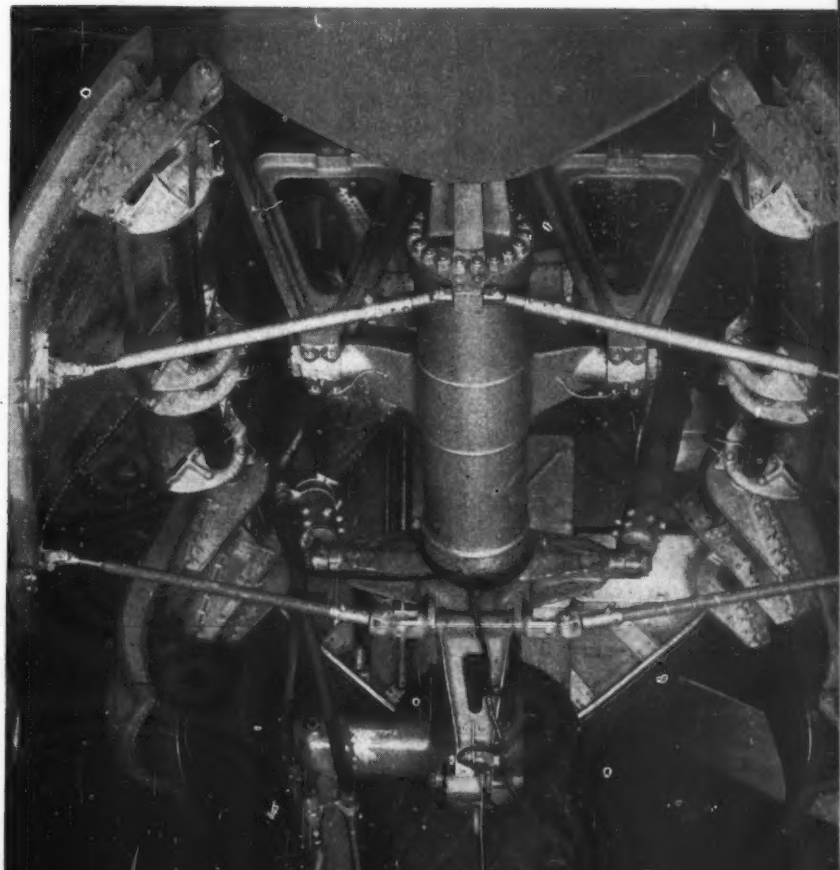


TABLE I

Low-Alloy, High-Strength Steels

Steel	Composition (per cent) C	Mn	P	S	Si	Cu	Ni	Cr	Mo	Zr	Al
Aldecor	0.12 max	0.15/0.40	0.08/0.15	0.05 max	0.35/0.75	0.35/0.60	....	....	0.16/0.28	....	....
Cor-Ten	0.12 max	0.20/0.50	0.07/0.15	0.05 max	0.25/0.75	0.25/0.55	0.65 max	0.50/1.25	....	....	....
Double Strength	0.12 max	0.50/1.00	0.04 max	0.05 max	....	0.50/1.00	0.50/1.00	....	0.10 min	....	....
Dynalloy	0.20 max	1.25 max	0.10 max	0.05 max	0.30 max	0.60 max	1.00 max	....	0.10 max	....	....
Hi-Steel	0.12 max	0.50/0.90	0.05/0.12	0.05 max	0.15 max	0.95/1.30	0.45/0.75	....	0.08/0.18	....	0.12/0.27
Mn-Ni-Cu	0.25 max	1.40 max	0.045 max	0.05 max	0.25 max	0.30/0.60	0.50/1.00	....	....	....	....
Mayari-R	0.12 max	0.50/1.00	0.08/0.12	0.05 max	0.10/0.50	0.50/0.70	0.25/0.75	0.40/1.00	....	....	....
NAX	0.08-0.15	0.50/0.75	0.04 max	0.05 max	0.60/0.90	....	....	0.50/0.65	....	0.05/0.15	....
Otiscoloy	0.15 max	0.90/1.40	0.08/0.13	0.04 max	0.10 max	0.30 min	....	....	....	....	....
Yoloy	0.15 max	0.60 max	0.05/0.10	0.05 max	....	0.75/1.25	1.50/2.00	....	....	....	....

point on the use of steels in designing all kinds of structures. As is well known, ordinary carbon steel was considered the standard material of construction until a few years ago at which time a series of low-alloy, high-strength steels were developed to save weight in steel structures. Utilization of these steels was brought to a standstill during the war, but since then successful application has increased to a remarkable extent, and these materials are now available in a variety of shapes. Compositions for the typical low-alloy, high-strength steels are presented in TABLE I.

Design Economy with Low-Alloy Steels

All these steels possess a minimum yield strength of 50,000 psi with a tensile strength in excess of 70,000 psi, which shows their great superiority over ordinary carbon steels for structural purposes. It is a proved fact that these steels have corrosion resistance superior to ordinary carbon steels under atmospheric conditions, their use having demonstrated that their service life under atmospheric conditions is approximately four times that of the carbon steels. These steels are little higher in price than the carbon steels, which means that they will in the future replace the latter steels in many structural applications because of the economy gained from greater strength and extended service life due to better corrosion resistance. Considerable iron can be preserved, and our best iron resources will not be depleted so rapidly. These steels have, therefore, become an important item and should receive serious consideration from designers of machines.

During the recent war there was not available enough of the different metals needed for the production of various alloy steels. As a result great effort was put forth to distribute the available metals such as chromium, nickel, molybdenum, etc., over a greater tonnage of steels to enable us to build the large number of machines and equipment required. A number of different types of steels were devised and used under the classification NE (National Emergency) Steels. Most all of these steels have been discarded because they were not quite suited for present-day applications. However, it has been demonstrated that the NE 8600 series steels have adequate properties for applications requiring a certain degree of hardenability. These steels are in the group now referred to as "H" steels, which enables the consumer to obtain maximum hardenability at minimum cost. The compositions of these steels will not be cited because they are easily obtainable in publications by the Society of Automotive Engineers and the American Iron and Steel Institute.

Another series of new low-alloy content steels has been developed containing additions of chromium, nickel and molybdenum, either separately or together. The percentages of these metals present in the steels have been held to a minimun, and the hardenability obtainable in the steels is assisted by a small boron addition. Maximum hardenability is thus secured with minimum alloy content. These steels have been given the designation "Superkore Steels."

SUB-ZERO SERVICE STEELS: During the past few years the use of oxygen gas has expanded greatly. In addition, a large development has taken place in the synthetic chemical field for the production of organic compounds. In the preparation of these products, fractionation of gases by temperature and pressure differentials has played an important role. The separation of the gases usually takes place at subzero temperatures, which makes it necessary for the equipment involved to operate at the low temperatures. Preservation of food at low temperature also requires equipment to operate under subzero conditions. Ordinary carbon and low-alloy steels lose toughness when exposed to subnormal temperatures. At a certain low temperature which is generally referred to as the transition temperature, a change occurs in their properties and these steels pass from ductile behavior to brittle behavior.

TABLE II

Effect of Temperature on the Toughness of 18-8 Steel (Type 304)

Test Temperature (degrees F)	Izod Impact Value (ft-lb)
70	119
32	120
— 58	117
—112	113
—300	119

Nickel steels containing 2 to 5 per cent nickel have been used successfully at temperatures down to minus 100 to 150 F, but at lower temperatures even these steels appear to lack adequate toughness. Work has been carried on to develop a still higher nickel content steel suitable for temperatures on the order of

TABLE III

Elevated-Temperature Fatigue Strength of Some Iron-Base Alloys

Alloy	Composition (per cent)									High-Temperature Strength (stress to cause rupture, 1000 hours, psi)		
	C	Cr	Ni	Co	Mo	W	Cb	Ti	N	1200 F	1350 F	1500 F
19-9 DL	0.26	18.95	9.05	.....	1.22	1.20	0.30	0.21	....	50,000	20,000	......
CSA*	0.38	18.52	4.55	.....	1.35	1.35	0.57	....	....	39,000	12,500	......
Timken	0.15	16.75	25.62	.....	6.29	....	....	....	0.13	42,000	19,000	......
S 495	0.48	13.68	20.06	.....	4.14	3.36	4.59	....	....	34,750	22,000	14,800
N 153	0.10	17.00	15.00	13.00	3.00	2.00	1.00	....	0.10	47,000	21,000	13,500
N 155	0.10	20.00	20.00	20.00	3.00	2.00	1.00	....	0.12	49,000	25,000	15,000
S 590	0.49	19.50	19.78	19.35	3.95	4.15	4.04	....	....	40,000	26,000	15,000
S 816	0.40	19.43	19.80	43.00	3.61	3.42	4.48	....	....	47,000	30,000	20,000

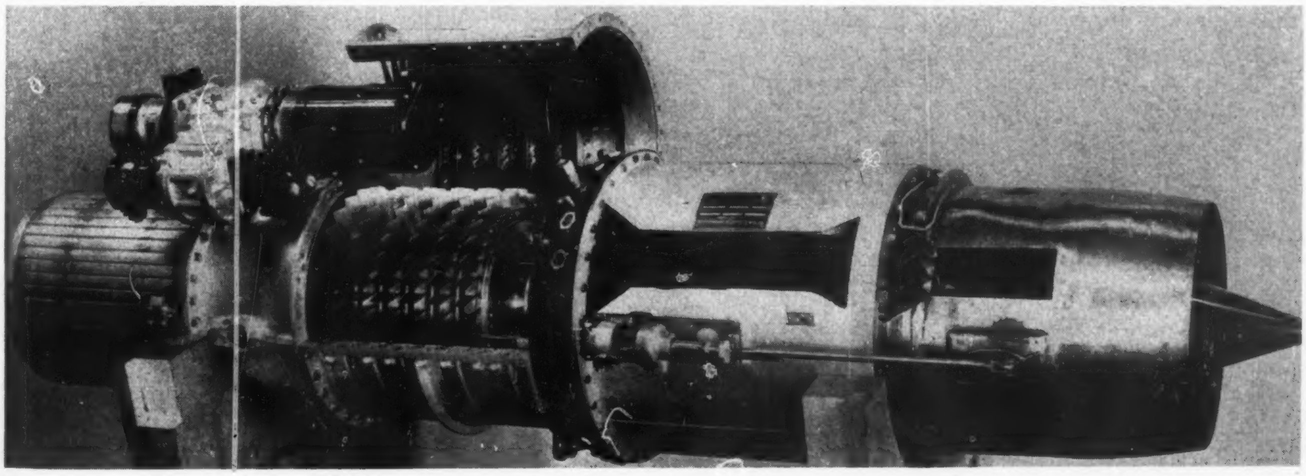
* This alloy contains 4 to 5% manganese.

minus 300 F. It is claimed that when the nickel content of a very low-carbon steel reaches about 8 per cent it will exhibit high toughness at minus 300 F in the heat treated condition. The use of this steel, however, is limited to the extent that it must be heat treated after welding if satisfactory toughness is to be retained at this low temperature.

HIGH-ALLOY CONTENT STEELS: The most widely known high-alloy content steels are the stainless steels which are represented by the plain chromium steels and the austenitic chromium-nickel steels. In the early development of stainless steels considerable effort was put forth to forge and hot-roll these materials into the standard articles being made from the ordinary carbon and low-alloy steels. Difficulties were encountered because of the greater strength of the stainless steels at high temperatures. Improvement in the hot-working characteristics of stainless, especially in the case of the austenitic chromium-nickel stainless steels has been continued, with the result that hot extrusion methods are now being commercially set up to produce almost any shape. This work will provide the machine designer with shapes that can be used to construct almost any kind of equipment that is required, and thereby give greater utility to the stainless steels.

Further progress in the fabrication of stainless steels has been brought about by application of the Heliarc process of welding. This process consists of welding either thin or relatively heavy sections of stainless steels with a metal arc under a protected atmosphere.

Fig. 2—Westinghouse Model 19B axial-flow gas turbine employs both stainless and super alloys in its construction



The outstanding attribute of the process is that metal can be deposited without introduction of carbon into the weld metal, which is one of the necessary requirements to preserve the high degree of corrosion resistance in the welds.

Stainless steels of either the chromium-nickel or plain chromium type have relatively poor heat conducting characteristics. This has been a handicap in applying the steels in applications requiring good heat transfer. It is now possible to produce a three-ply stainless article with copper representing the middle lamination, which acts to greatly improve the heat conductivity. This material can be employed in various ways, and should provide the means for overcoming the relatively poor heat transfer characteristics of these steels.

Field for Stainless Increased

During early development of stainless steels it was recognized that carbon was unquestionably one of the most influential elements. Carbon influences both mechanical properties and corrosion resistance of the steels. In the case of the austenitic nickel steels, carbon makes them subject to intergranular corrosion and under certain conditions of temperature and corrosion the steels are badly attacked during operation. The susceptibility of the austenitic chromium-nickel steels to this defect greatly limited their utility during early development, but it subsequently was found that the trouble could be practically eliminated by the introduction of columbium and titanium. The columbium addition has been

TABLE IV

Elevated-Temperature Fatigue Strength of Some Nickel-Base Alloys

Alloy	Composition (per cent)									High-Temperature Strength (stress to cause rupture, 1000 hours, psi)		
	C	Cr	Ni	Co	Mo	W	Cb	Ti	Al	1200 F	1350 F	1500 F
Refractaloy 26	0.03	17.90	37.00	20.00	3.00	....	....	2.99	0.58	38,000	.....	.....
Modified Inconel	0.03	14.40	74.82	.....	...	....	....	2.77	0.63	38,000	16,500	.....
Nimonic 80	0.04	21.18	74.23	.....	....	....	....	2.44	0.63	45,000	29,000	7,500
K 42	0.06	18.00	42.00	22.00	....	....	....	2.56	0.59	38,000	28,000	11,000
Inconel X	0.05	15.00	73.00	.....	....	....	1.00	2.50	0.70	41,000	.....	17,000
Hastelloy B	0.05	.....	63.00	.....	28.00	....	....	....	....	40,000	26,000	10,700
Hastelloy C	0.10	15.00	58.00	.....	16.5	5.00	....	....	....	.....	.....	10,700

found to have wider application because it can be used in steels for welded construction both as parent metal and as weld metal if approximately ten times as much columbium as carbon is maintained in the steel. Even in columbium-bearing steels of the 18 per cent chromium-8 per cent nickel type it has been found advisable to hold the carbon at a maximum of 0.07 or 0.08 per cent.

Molybdenum bearing 18-8 steels appear to have the best overall resistance to corrosion of any of the stainless steels. These steels, however, are also subject to intergranular corrosion and in the attempt to eliminate this defect, development work has been in progress on 18-8 steels containing both columbium and molybdenum. The results have shown that steels within the following specifications have highly useful properties.

Carbon	.07% max.	Molybdenum	2.00%- 2.50%
Manganese	1.50% min.	Columbium	.60%- 1.00%
Silicon	.75% max.	Sulphur	.03% max.
Nickel	13%-15%	Phosphorus	.03% max.
Chromium	17.75%-19.50%		

One of the marked advantages these steels have is that equipment can be fabricated by welding from annealed plate of the steel, after which the vessel or other type of part can be given a simple stress relieving heat treatment which does not cause the metal to suffer a serious decrease in resistance to corrosion. Another feature of this steel is that it has relatively high strength at elevated temperatures.

As previously stated, considerable interest has developed during the past few years in the use of metals at extremely low temperatures. It may not be generally known and it should be emphasized that the 18-8 steels either of moderately high or low-carbon content retain their toughness practically unimpaired at temperatures down to minus 300 F and lower. The steels also remain ductile at these low temperatures and, unlike ordinary steels, do not have a transition temperature within the ranges mentioned. This behavior is considered unique in the field of ferrous metallurgy. For the purpose of illustration a few data are presented in TABLE II. In addition, welds of the 18-8 steels have high toughness at low temperature. Thus, these steels are ideal materials for low-temperature service involving even the handling of liquid gases such as oxygen and nitrogen.

SUPER ALLOYS: Probably the most important series of new metals are the super alloys developed for military use during the war. These alloys were devised for application in gas turbines for aircraft, *Fig.* 2. Initially the work was started on the alloys in view of aircraft turbosupercharger requirements. These turbosuperchargers, as well as the subsequently developed gas turbines, operate at higher temperatures than the steam turbine. There are a number of ramifications to the gas turbine principle, including aircraft jet engines, the marine gas turbine for ship propulsion, gas turbines for railroad engines, and aircraft ram jets, all of which utilize these alloys that must retain their dimensions to a close degree during operation at temperatures of at least 1200 F.

Metals developed for this purpose may be divided into two broad groups, the first of which represents high-temperature metals capable of being hot worked

Fig. 3—Westinghouse 2000-hp gas-turbine generator has a forged rotor of stabilized 19-9 stainless steel and blades of precision-cast cobalt-chromium-tungsten alloy

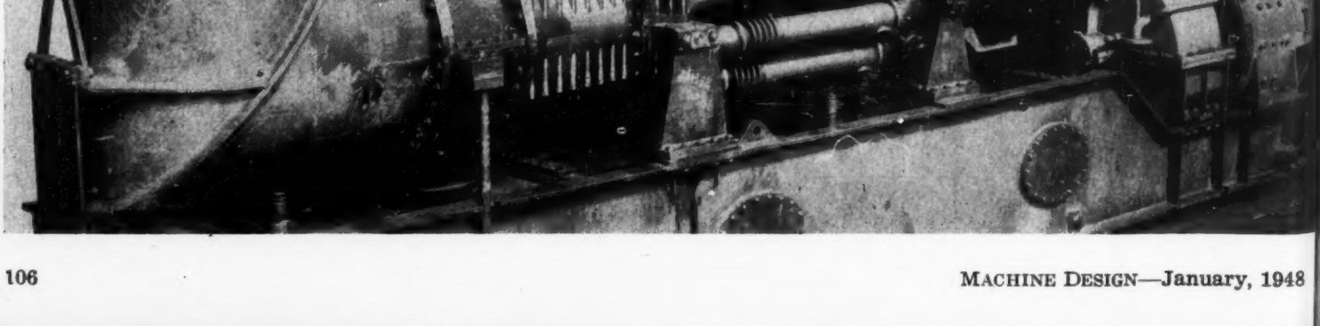


TABLE V

Elevated-Temperature Fatigue Strength of Some Cobalt-Base Alloys

Alloy	Composition (per cent)							High-Temperature Strength (stress to cause rupture, 1000 hours, psi)			
	C	Cr	Ni	Mo	W	Cb	Co	1200 F	1300 F	1500 F	1600 F
HS 21 (Vitallium)	0.28	27.00	2.00	6.00	....	....	Bal	44,000	27,000	15,000	13,000
HS 23 (61)	0.40	26.00	1.00	....	6.00	....	Bal	47,000	30,000	21,800	12,000
HS 27 (6059) ..	0.40	26.00	32.00	6.00	....	....	Bal	46,000	29,000	18,400	12,000
HS 30 (422-19) .	0.40	26.00	15.00	6.00	....	....	Bal	.....	.....	21,700	14,800
HS 31 (X 40) ..	0.50	25.00	11.00	....	7.5	....	Bal	46,000	39,000	23,400	18,000

into forms such as plate, sheet, strip, bars, etc., whereas the second series represents metals which cannot be hot worked and which consequently are used primarily in the form of castings. These high-temperature alloys may be further subdivided in accordance with the base from which they are derived, the three bases being iron, nickel and cobalt.

No attempt will be made to describe all of the properties of the alloys. Their compositions with the exception of the small amounts of silicon and manganese they contain, will be given with data relating to high-temperature strength. This information should give the designer a good idea of what may be expected of the materials. The iron base alloys are thus described in Table III.

It is observed from the data of Table III that the total alloy content of the iron base aloys vary from approximately 35 to 70 per cent. In all of these high-temperature alloys the metal chromium represents one of the chief constituents. Chromium is necessary to impart resistance to scaling at the high temperatures, and is assisted by nickel, cobalt, molybdenum, tungsten, columbium, etc., to promote improved strength at the high temperatures.

The second series of super alloys, representing the nickel-base metals, is in like manner described in Table IV. It is again noticed that the metals tungsten, molybdenum, columbium, titanium and aluminum are utilized in imparting strength at high temperatures. The pertinent difference between the nickel-base alloys and the iron-base alloys is the fact that the high-temperature strength is inherently present in the iron-base alloys whereas the high-temperature strength of the nickel-base alloys is primarily developed as a result of precipitation hardening, which is brought about by special heat treatment applied prior to application in service. This heat treatment, consisting of heating at some temperature between about 1200 and 1500 F for a few hours, is applied to parts prior to placing them in service. The question often has been asked as to which type of alloy is the better for a given application. The answer to this question can be obtained only by service experience, which now is being developed.

Cobalt-Base Alloys Require Casting

The third series of high-temperature alloys represents the cobalt-base metals as described in Table V. Cobalt-base alloys have been used primarily in the form of castings, but there is little doubt that as time goes on successful methods will be developed for hot working them. The metals tungsten, molybdenum, and columbium also have been incorporated either separately or together in the cobalt-base alloys to impart high-temperature strength. These alloys have been found to possess excellent casting properties and have been used widely as precision parts made by the investment casting process. The alloys are not so readily machined as some of the other high-temperature alloys, but they can be shaped to close dimensions by limited machining and grinding.

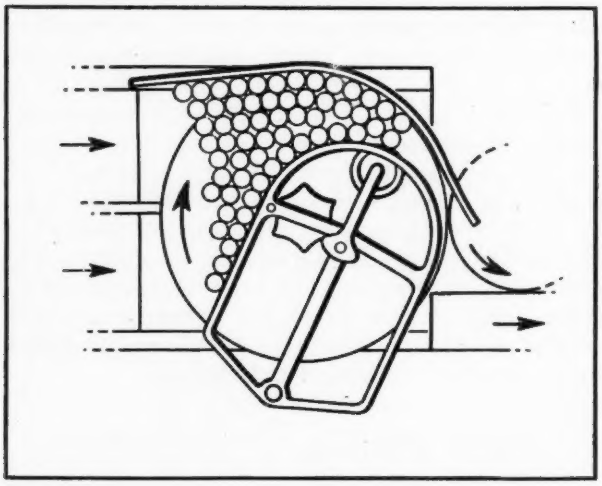
These super alloys have already been used in thousands of turbosuperchargers, and large numbers of jet aircraft engines. Also, more recently, they have been used in a gas-turbine generator set, *Fig.* 3. Gas turbines for railroad engines have been constructed using the super alloys, and the predictions are that within a few years there will be a heavy demand for these power units. In the offing are gas-turbine electric locomotives less than half the length and two-thirds the weight of present-day diesel-electrics.

Powder Metallurgy Obviates Melting

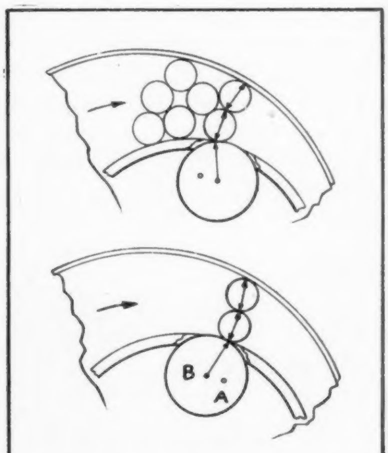
PURE METALS: Progress has been made on the commercialization of the pure metals molybdenum, tantalum and titanium. Work with molybdenum and tantalum has consisted of developing processes for producing the metal in ingot form and hot rolling the ingots into stock forms such as bars, rods, sheets, and strip. This work has been done on mills used in steel production. Heretofore only small sections of molybdenum have been available which has greatly retarded the use of the metal. Indications are that it will be possible to produce fairly large wrought sections of molybdenum, which will immediately be used in chemical equipment to resist corrosion because of the inertness of the metal in contact with corroding media of a reducing nature. Likewise, tantalum is finding many applications where its extremely high resistance to chemical attack is of value. Its main limitation, however, is rapid oxidation at elevated temperatures.

The Bureau of Mines recently announced in Information Circular 7381 entitled, "Metallic Titanium and Its Alloys" that a process has been developed to produce ductile titanium metal. Ductile titanium has been hot rolled into bars, rods, sheet, etc., and exhibits good corrosion resistance and mechanical properties. Titanium is a relatively light metal with a fairly high modulus of elasticity (15,000,000 psi) which, with the accompanying resistance to corrosion, makes the wrought metal ideally suited for light-weight, high-strength structures.

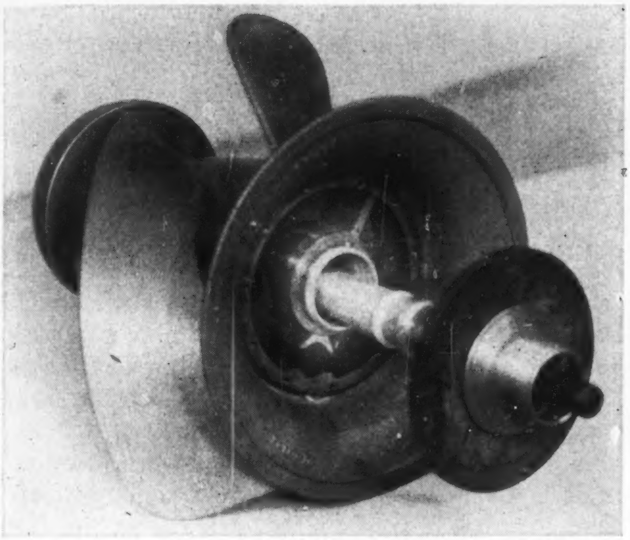
Scanning THE FIELD *for Ideas*

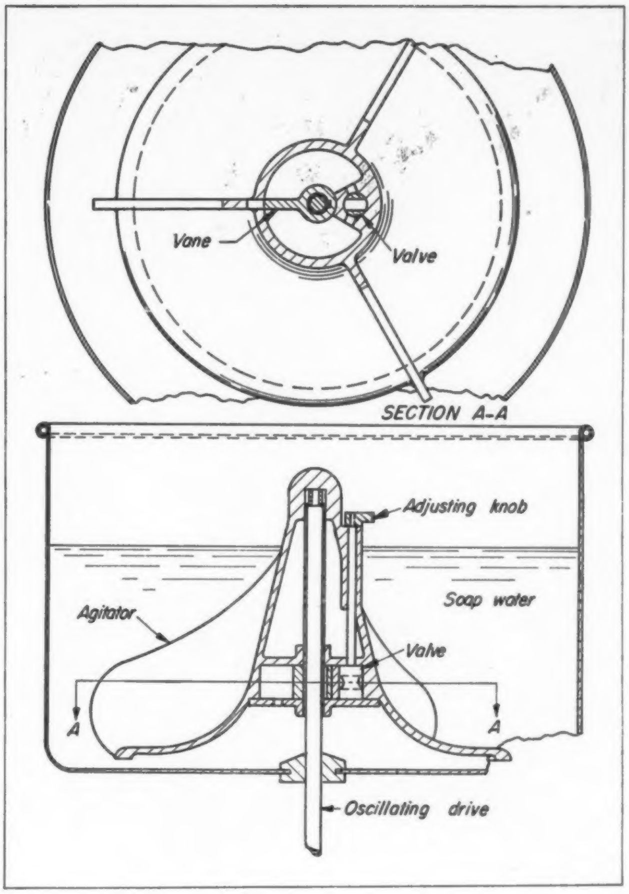


Unscrambler, below, for glass containers as they are fed into a bottling machine utilizes an oscillating free wheel to automatically prevent jams from forming in the throat of the conveyor line. The schematic sketch, left, illustrates the action of the wheel. Having a 5/8-inch stroke governed by a cam, the oscillator arm keeps the containers in a constant, gentle motion. This action frees containers about to enter the throat. The free wheel turns against the bottles at the jamming point and releases them, avoiding damaging impact. This wheel together with two bottles in jamming position makes an easily broken arch of three as shown in the simplified sketch below. The motion of the wheel frees the containers so that they may enter the conveyor one at a time. Octagonal-shaped bottles are also unscrambled by this oscillating arm and wheel arrangement designed by the Hartford Empire Co.



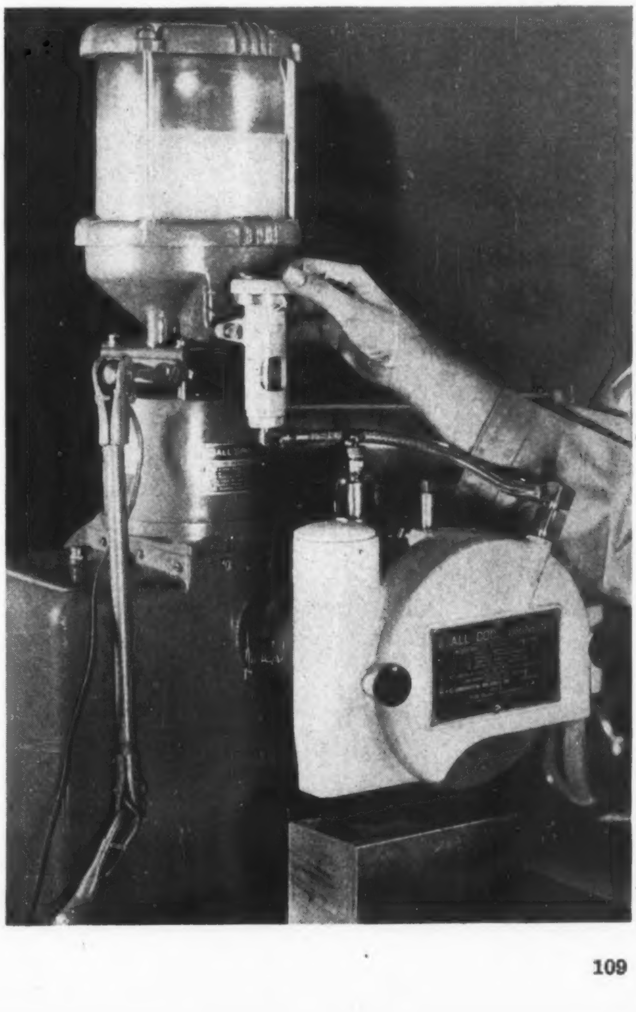
Adjustable-speed and variable stroke for an agitator having an oscillating drive are



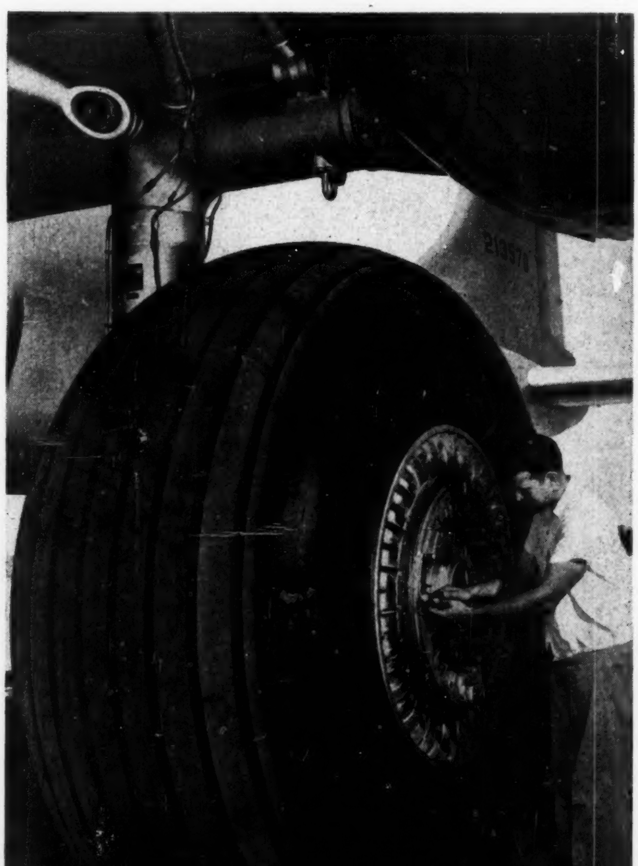


obtained through a novel fluid coupling. Applied to a washing machine agitator for controlling the washing action, the coupling is shown in a partially disassembled view, above, and in sectional drawing, right. Action of the coupling, invented by Veb Hanson, Fairfield, Conn., is controlled by the position of a valve which controls the passage of fluid from one side of the coupling to the other.

Agitator is not direct driven. Instead a vane, driven by the oscillating drive shaft, is the driving member of the hydrostatic coupling. When the valve in the coupling is closed, the agitator moves virtually at the speed of the oscillating shaft. When the valve is full open, action is a minimum. Fluid in the coupling is the same as that in the tank and enters through a small port in the bottom plate of the coupling. This port is so small that when the vane oscillates not enough fluid can be expelled to affect normal operation. Coupling vane is affixed to a tube driven by the squared end at the top of the drive shaft.



Coolant for grinder, right, is introduced through the grinding wheel arbor to effect cooling at the point of contact with the work, yet have operator visibility comparable to dry grinding. Developed by The DoALL Co., the grinder does not require splash guards, settling tanks and pump motor. Mounted on the spindle column, a reservoir feeds the coolant through a sight drip valve at the rate of one to four drops a second. Directed to the front of a wheel adapter, the coolant enters a hole in the arbor of the wheel. Since the wheel has no lead or ceramic core the coolant enters the wheel and is thrown by centrifugal force to the outside grinding face. Because the coolant flows through the wheel it flushes the surface, resulting in longer wheel life and better finishes especially on the



softer materials. Also, the grinding temperatures are relatively low, reducing the formation of surface cracks on the work.

Four-wheel landing gear, left, on B-36 superbombers built at Consolidated Vultee Aircraft Corp., replaces the single-wheel gear, left below, used on the experimental XB-36. The eight 56-inch wheels distribute the bombers 278,000 pounds over a greater area than the two 110-inch wheels on the experimental model. This improved weight distribution, coupled with shortened landing runs resulting from reversible-pitch propellers, will enable the bomber to utilize a greater number of airfields. The truck type gears are also lighter, are safer in event of blowout and have easier-to-service tires. Dual wheels are utilized on the nose landing gear.

Aluminum alloy burner, below, used by the heater division of McCulloch Motors in their gas space heaters is the first aluminum burner approved by the American Gas Association. A series of fins provide sufficient heat transfer to prevent the burner from reaching a dangerous temperature. Concentration of fin area near the mixing end of the burner is provided because the highest burner temperatures are reached when flashback occurs and burning takes place inside the burner. This always happens near the inlet end.

Air Jets

. . . their characteristics for gaging devices

By R. S. Elberty
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AIR jets as used for ultrasensitive gages are not new, some of the foreign patents dating back to 1928. Today, air gages are in common use in precision manufacturing, being particularly successful for gaging bores. Air jets controlling electrical switching have been applied to machine tools as automatic sizing devices, making it possible to measure the work while it is being machined without the measuring element touching the work. Since the use of air jets for gaging will undoubtedly expand further, design engineers will be interested in actual characteristics of these devices, as covered by the test results discussed in this article.

FUNDAMENTAL CIRCUIT: *Fig.* 1 shows the basic circuit for an air gage in which pressure is used for indication. If P_1 is constant, the pressure P_2 will vary as d is varied. For $d = 0$, $P_2 = P_1$ or a general equation might be written

$$P_2 = P_1 \frac{r_2}{r_1 + r_2}$$

Since the measurement d affects r_2, then P_2 can be used as an indication of d. The pressure drop through a smooth pipe will vary as the square of the volume of air passing through the pipe and r_1 and r_2, the resistances in the foregoing equations, are variables.

The pressure drop across an air jet 0.250-inch long and 0.026-inch diameter is shown in *Fig.* 2 giving an indication of the variation of r_1 and r_2 in the usual air gaging operation.

FACTORS GOVERNING SENSITIVITY: If

$$P_2 = P_1 \frac{r_2}{r_1 + r_2}, \text{ then } \frac{dP_2}{dr_2} = P_1 \frac{1}{r_1 + r_2}$$

or the amplification is equal to $P_1/(r_1 + r_2)$. When r_2 is infinite, the amplification is infinite. However, in the actual testing of air jets, it was noted that when d was varied the maximum amplification was obtained with P_2 about sixty to seventy per cent of P_1, a typical characteristic curve being as shown in *Fig.* 3. This figure shows that the air jet is practically shut off for all small values of d, and r_2 did not begin to change greatly in the case of the jet tested until d was equal to 0.0015-inch. As the equation indicates, the point of maximum sensitivity can be shifted by a variation of r_1 and a needle valve was used for r_1 so that a maximum sensitivity could be determined for each of the jets tested.

In *Fig.* 4 is shown a group of tests made with a jet diameter of 0.050-inch, varying the settings of the needle valve to modify the value of r_1. With the needle valve being progressively closed (r_1 progressively increased) the amplification increases to a

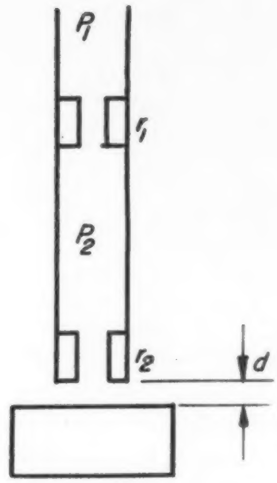
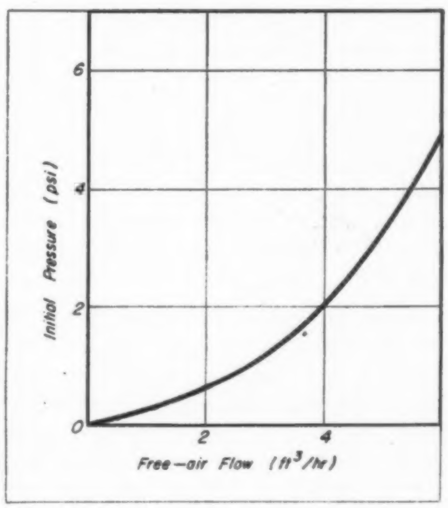


Fig. 1—Top—Basic circuit for pressure-type air gage. The orifice r_1 is usually smaller than r_2. There is some advantage in making r_1 adjustable

Fig. 2—Below—Pressure drop across an air-jet. The jet in question was 0.026-in. diameter by 0.025-in. long. The pressure drop varies as the square of the volume where the velocities are below sonic



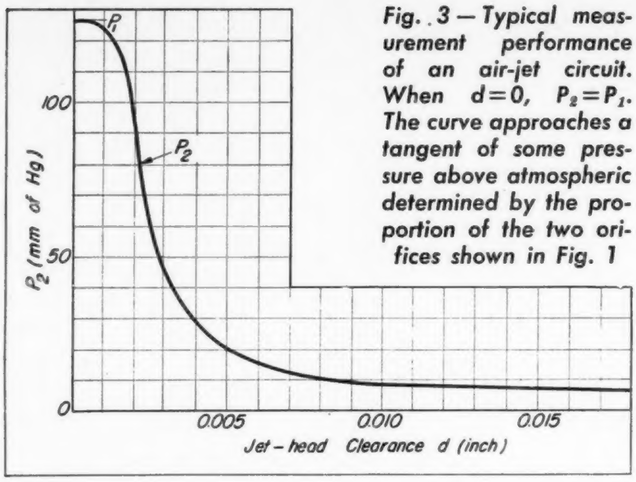
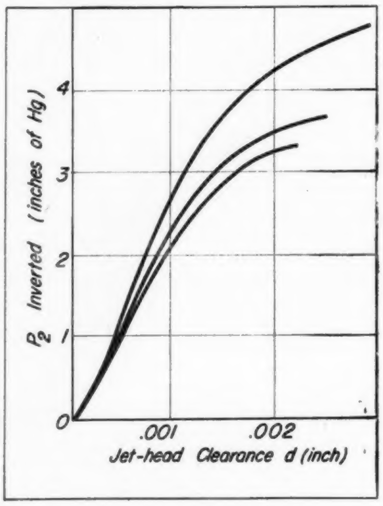


Fig. 3 — Typical measurement performance of an air-jet circuit. When $d=0$, $P_2=P_1$. The curve approaches a tangent of some pressure above atmospheric determined by the proportion of the two orifices shown in Fig. 1



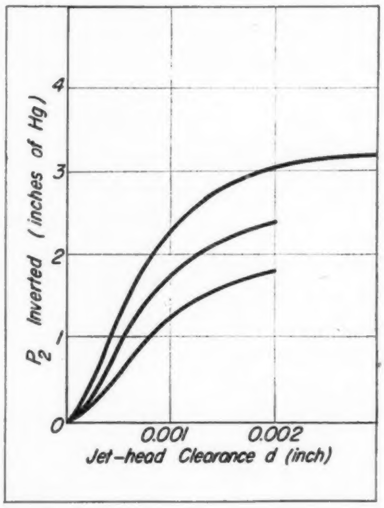


Fig. 4—Above Left—Effect of variation of orifice r_1. In this case, a needle valve was used for r_1, different amplifications being obtainable by adjustments of the needle valve. P_2 is shown on an inverted scale

Fig. 5—Above Right—Effect of variation of P_1. Amplification varies directly as P_1, the test performance verifying mathematical analysis

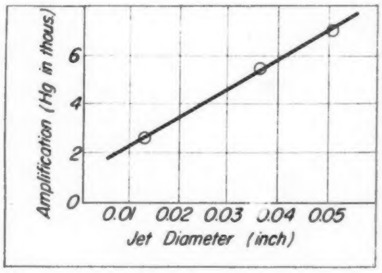


Fig. 6—Above—Effect of jet diameter on amplification. Sensitivity of an air-jet gaging device can be increased by increasing the jet diameter. This curve shows the maximum sensitivities that were obtained by adjusting the needle valve to make r_1 the optimum valve

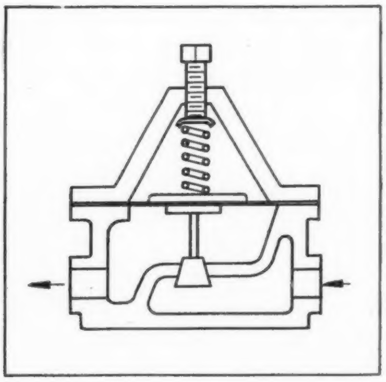


Fig. 7 — Right—Diaphragm-type pressure regulator. Regulation is obtained by the balancing of pressures on both sides of the diaphragm member

maximum and then decreases again. Since the optimum value of r_1 can be set readily for any condition, further tests were made with the needle valve set to obtain the best amplification characteristics.

The formula also indicates that sensitivity or amplification is affected by the initial pressure P_1. In *Fig.* 5 are plotted results of a group of tests made with a jet diameter of 0.012-inch and with initial pressures of 10, 15, and 20 psi for P_1. These tests show that the sensitivity is proportional to P_1. The air pressure may be varied over a large range in actual practice. For example, the Landis-Solex sizing device used on Landis grinding machines uses a pressure P_1 of about 2 psi and the Pratt and Whitney Air-O-Limit plug gage uses a pressure of 30 to 35 psi.

LINEAR CHARACTERISTICS: Since the pressure characteristic follows a curved line, comparison between various jets would be difficult unless some standard point for comparison were determined. *Fig.* 3 shows that the curve changes from concave downward to concave upward. At this point of change, sensitivity is greatest and the characteristic may be considered a straight line. The variations from the straight line may be shown from the accompanying tabulation comparing results from two tests made under different conditions.

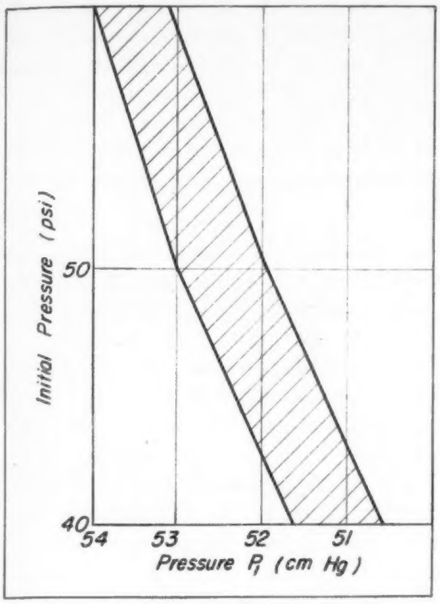
Linear Characteristics of Air Gage

Test Condition	Jet Size (inch)	Pressure P_1 (psi)	Measured Sensitivity (in. Hg/in.)	Extent of Linear Effect (inch)	Deviation From Linear Effect (inch)
A	0.022	11½	1875:1	0.0007	0.000017
B	0.050	9	3500:1	0.0007	0.0000178

For practical considerations, an air gage at a P_1 of 9 to 15 psi might be considered as having a linear characteristic over a pressure range of about 3 inches of Hg. Translated into terms of measurements, the sensitive gages will have a narrower effective linear range than those with less sensitivity.

JET DIAMETER: When the maximum sensitivity of various air jets is compared, the greater sensitivity is obtained with the large diameter jets, sensitivity being roughly proportional to the jet diameter. *Fig.* 6 shows a comparison of sensitivities obtained with 0.012, 0.035 and 0.050-inch diameter jets at a P_1 of 9 psi. This is also shown in a comparison of *Figs.* 4 and 5.

SUMMARY OF JET TESTS: Review of the foregoing discussion discloses that



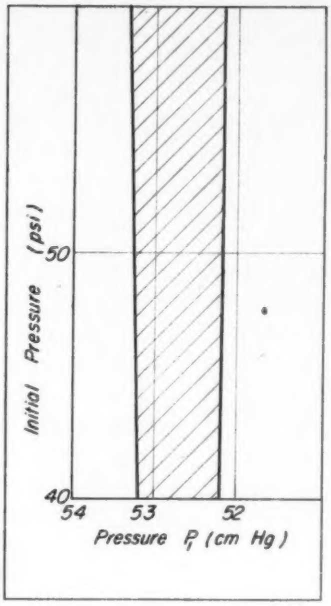


Fig. 8—Above Left—Performance of commercial pressure regulator. A drop in initial pressure is accompanied by a drop in the regulated pressure. Test was made with a small volume of air passing through the regulator

Fig. 9—Above Right—Two regulators connected in series. The faulty performance shown in Fig. 8 is corrected by using two regulators

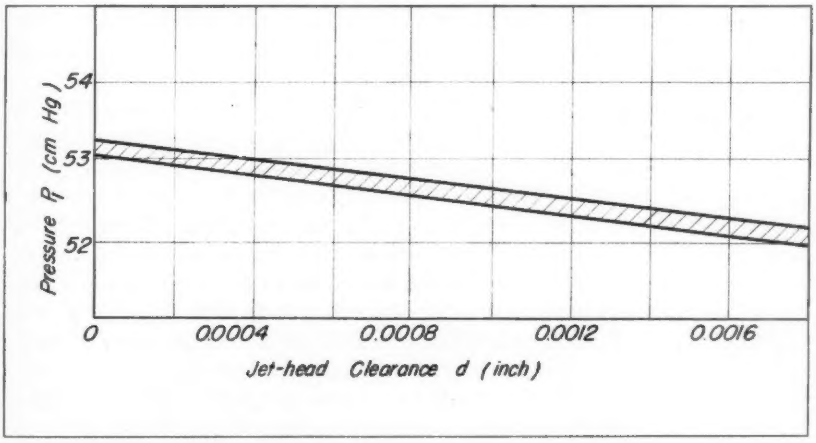
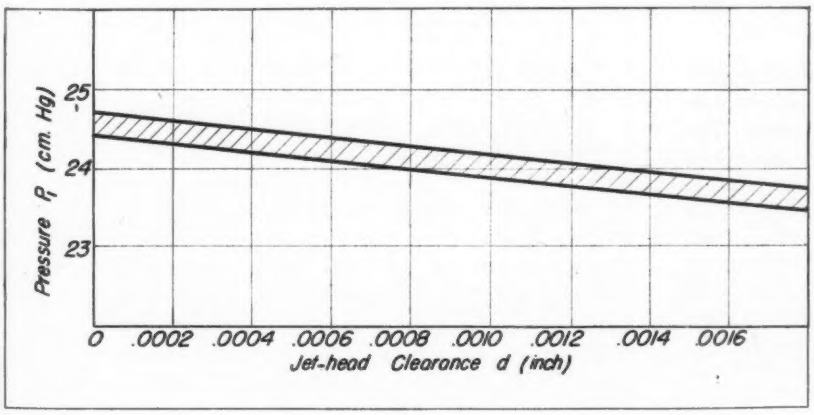


Fig. 10—Above—Effect of volume of air on commercial regulator. The pressure drops as volume is increased; linear characteristic is good and enables the regulator to be used for gaging operations

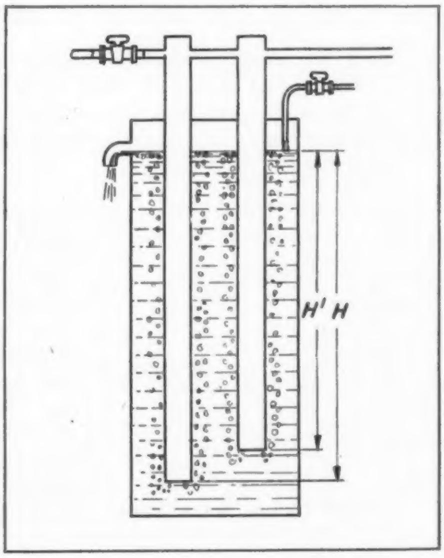
Fig. 11—Below—Two regulators in series. Addition of the second regulator does not correct effect of volume of air on regulated pressure



1. Pressure drop across an air jet varies as the square of the velocity of air through the jet (below sonic speeds)
2. Sensitivity of a pressure-type air gage varies directly as the initial pressure P_1
3. Sensitivity of an air jet varies directly as the diameter of the jet
4. Zone of greatest sensitivity of a pressure-type air gage may be considered as a linear function of the measurement over a pressure range of 3 inches of Hg for gages using a P_1 of 9 to 12 psi.

PRESSURE REGULATORS: In the analysis of air gages, it was shown that the calibration of the gage is affected by the pressure, P_1. It is therefore essential to start any air gaging operation with a source of constant air pressure—a difficult requirement to achieve. *Fig.* 7 shows a schematic view of a diaphragm type of pressure regulator. The valve is held open by the spring until the discharge pressure acting against the diaphragm is sufficient to close the valve. The valve has a surface exposed to the initial and secondary pressures. Also, the valve is not perfect and tends to leak, requiring greater pressures to seal against high initial pressures. It is therefore apparent that a change in the initial pressure affects the calibration of the regulator. *Fig.* 8 shows the per-

Fig. 12—Below—Water-column pressure regulator. In this regulator, the stopcock is adjusted so that a small amount of air in excess of the requirement is allowed to bubble through the column of water of a fixed height



formance of a commercial regulator of the diaphragm type when operated under various initial pressures. With two regulators connected in series, the performance is improved, *Fig.* 9.

The diaphragm type of regulator has another defect. As the volume of air increases, the valve must open further. This is accompanied by an extension of the spring and a reduction in the secondary pressure to balance the extended and weakened spring. This effect is shown in *Fig.* 10. This disadvantage cannot be reduced by connecting two regulators in series, *Fig.* 11 showing the results of tests using this circuit.

The Solex type regulator is based on the pressure of a fixed column of water. It is used on the Federal Metricator gage, the regulator being built-in as a part of the gage. *Fig.* 12 shows a schematic diagram of this type of regulator as taken from one of the Mennesson patents. The Landis Co. regulator is a relief valve, accurate regulation of which is dependent on the volume of air being greatly in excess of the requirements. It would be subject to the same errors as the diaphragm regulator shown in *Fig.* 7 if the original volume of air were varied. However, the Landis device insures a constant volume of air through a constant-displacement compressor driven at a constant speed.

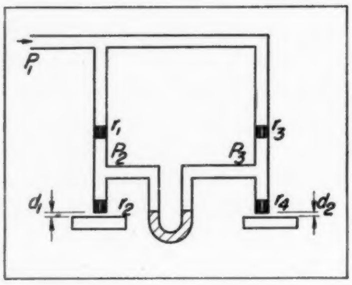


Fig. 13—Left—Balanced-bridge network of air jets. This is the Wheatstone bridge applied to air circuits to balance out variations in P_1

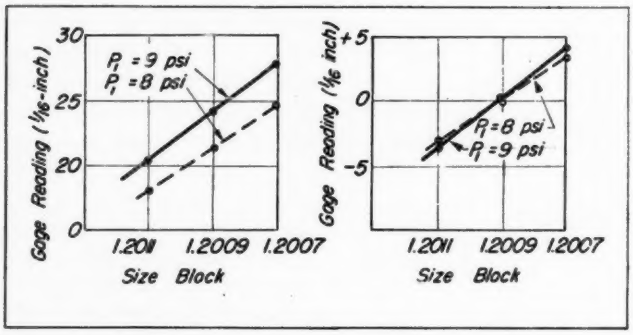


Fig. 14—Above Left—Effect of variation of P_1 in simple gage. The circuit as shown in Fig. 1 is not suitable for precise gaging if P_1 varies as this illustration shows

Fig. 15—Above Right—Effect of variation of P_1 in balanced gage. The circuit shown in Fig. 13 will give accurate readings for wide changes in P_1 provided the gage is balanced at the center of the limits to be measured

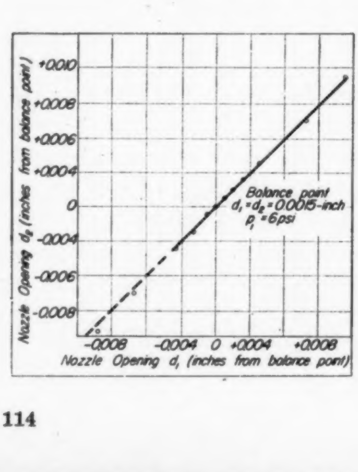


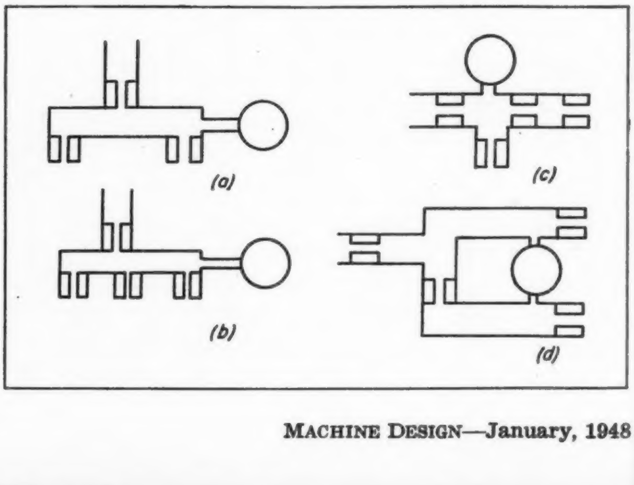
Fig. 16—Left—Comparison of two air jets with balanced circuit. The two jets to be compared were made as closely alike as possible. Actual test points vary from the theoretical curve slightly, showing that the jets were not completely matched. The circuit used is shown in Fig. 13

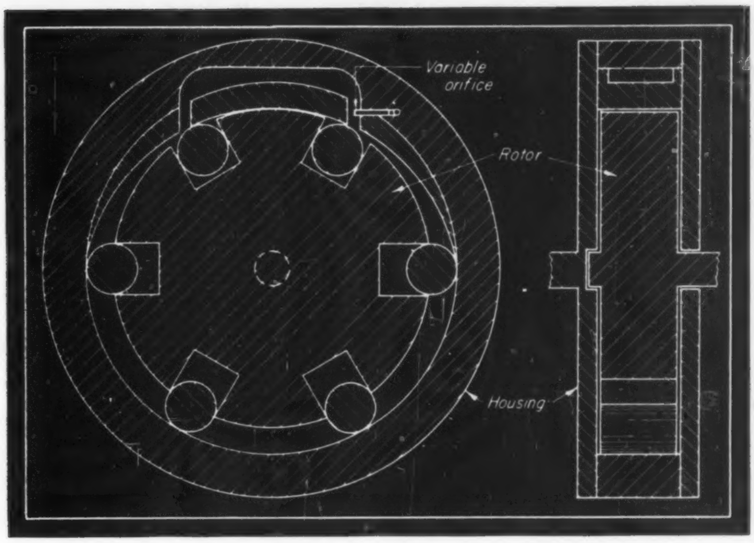
Circuits For Air Gages: In order to reduce the effect of pressure variations in air gaging operations, it is possible to use a combination of jets connected in a bridge circuit. *Fig.* 13 shows a grouping of air jets in the well known Wheatstone bridge circuit used in electrical measurements. If $r_1/r_2 = r_3/r_4$, then the bridge is balanced and $P_2 = P_3$. For any change in P_1, there is a corresponding change in P_2 and P_3, but P_2 and P_3 will still be equal. Needle valves may be used for r_1, r_3 and r_4 and the gage can be balanced for any condition of the r_2 jet and d. At this point of balance, changes in P_1 will have but a slight effect on the calibration of the gage. *Fig.* 14 shows the effect of a pressure, P_1, variation of 12 per cent on a standard gage and *Fig.* 15 shows the effect of this same variation on a gage using a balanced Wheatstone bridge circuit. In these examples, the error would be -0.0002 to -0.00015-inch with the unbalanced circuit, and -0.000030 to $+0.000025$-inch with the balanced circuit.

The Wheatstone bridge circuit could be used for comparing or matching two measurements or it could be used for automatically controlling a machine for selective sizing operations. *Fig.* 16 shows actual test results comparing the opening of two nozzles (d_1 and d_2) with the network connection shown in *Fig.* 13.

The use of a bridge circuit suggests that other complex circuits could be devised. There is an analogy to electrical circuits which will assist the designer in selecting a circuit to fit a given set of circumstances. Calculations are difficult since the pressure drop across a resistance varies as the square of the volume of air through the resistance, the electrical formula $E = IR$ becoming $P = V^2R$ and other formulas and calculations are complicated by this nonlinear equation. *Fig.* 17 shows a number of network circuits for air jets.

Fig. 17—Complex air network circuits. Circuits a and b are averaging circuits used in plug gaging; c and d may have some practical use, and are included to illustrate the possibilities of air network circuits





Hydraulic Clutch

. . . of the positive-displacement type is analyzed, furnishing essential design information and providing a basis for the evaluation of clutch performance

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INHERENTLY smooth operation and precise control possible with hydraulic units have from time to time inspired engineers to design clutches based on hydraulic principles. An example is the type of cluth discussed and analyzed in this article, which utilizes a positive-displacement pump as the principal element. As far as is known, there has been published no analysis of the mechanics of operation of this type of clutch.

The fact that the performance of some clutches of this type has not been up to the desired standard led to the development of the analysis presented in this article. It may be used as the basis of the design of such mechanisms and will be of considerable assistance in interpreting experimentally determined performance characteristics and correcting possible errors in design.

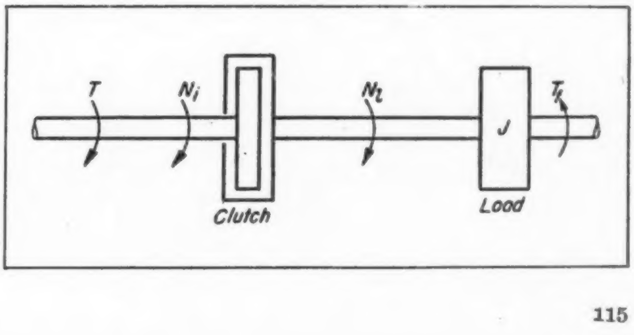
An elementary analysis will be developed on the basis of considering the picking up of a load, consisting of inertia and constant friction, by means of a positive displacement pump, the performance of which is described in terms of its displacement and slip characteristic. A simple valve mechanism and mode of operation are assumed, *Fig.* 1, in order to simplify the mathematical analysis although more complicated valve operations may be considered within the general framework of the analysis which follows.

It is found that while smooth operation of this type of clutch is definitely possible it is highly probable that few if any commercially available models take full advantage of the inherently good characteristics of the device. This is due to the fact that operational characteristics are critically dependent upon the valve operation, and in such a manner that intuitive ideas are in direct contradiction to the facts. One is led, therefore, if guided by intuition, to design a valving cycle which may very well defeat the primary purpose of the device, which is smooth pickup of load.

ANALYSIS: In order to simplify the analysis of the mechanics of operation the idealized setup illustrated

Fig. 1—Top—Elementary design of positive-displacement pump modified to permit operation as a clutch

Fig. 2—Below—Schematic diagram of clutch and connected load having inertia and friction



in *Fig.* 2 is used as the basis of the development. A rigid shaft rotates at constant angular speed N_i being driven by a motor which exerts torque T to maintain the motion. This shaft is rigidly connected to the rotor of a positive-displacement hydraulic pump. The housing of this pump is connected to a rigid shaft whose axis coincides with that of the driving shaft. This shaft is in turn connected to the load consisting of a mass whose moment of inertia with respect to the axis of rotation is J. Frictional drag exerts a constant torque T_f on the load shaft in a direction opposing the rotation. The angular speed of rotation of the load shaft is N_l.

Discharge port of the pump, shown in *Fig.* 1, is connected through a conduit, in which there is a variable orifice, to the intake port of the pump. The area of the orifice is subject to variation by means of a suitable control mechanism.

As the input shaft rotates at the speed N_i the rotor of the pump turns relative to the housing at speed $N = N_i - N_l$. If the load is initially at rest, it will remain at rest until the torque on the load shaft is sufficient to overcome the frictional drag T_f. The torque transmitted by the clutch may be discussed in terms of the characteristics of a positive-displacement pump.

The performance of a positive-displacement pump may be described in terms of two equations:

$$Q = DN - C_s P \qquad (1)$$

$$T = PD + KN \qquad (2)$$

where Q is the delivery of the pump in units of volume per unit time; D is the displacement of the pump in units of volume per unit rotation of the shaft; C_s is the slip in units of volume per unit time for unit pressure difference; P is the pressure difference between discharge and intake of the pump in units of force per unit area; T is the torque required to drive the pump in units of force times distance; and K is a coefficient of the pump and fluid which determines the torque due to viscous drag.

For the purpose of this discussion the coefficient K will be assumed to be negligibly small. Hence the torque equation for the pump is

$$T = PD \qquad (3)$$

Flow through the variable orifice may be described by the equation

$$Q = CAP \qquad (4)$$

where C is the orifice discharge coefficient and A is the area of the orifice, providing the viscosity is constant.

Before proceeding to the use of these equations in the description of the motion of the system the physical significance of each will be considered.

The rate Q at which liquid is delivered by the pump is described by Equation 1. The positive displacement pump is characterized by the constancy of its delivery at a given speed, diminished only by the slip which is proportional to the pressure. Each unit of angular motion of the shaft displaces a quantity of liquid D, hence in N radians the quantity DN is displaced and if N radians displacement occurs in unit time the delivery is at the rate DN units of volume per unit time. Since the seals between surfaces which move relative to each other are not perfect, some liquid flows from the discharge side of the pump back to the intake side at the rate $C_s P$.

Torque required to drive the pump is directly proportional to the pressure and the displacement of the unit as shown in Equation 2. Neglect of the term KN is justified in the present development wherein inertia and friction torques far outweigh the viscous torque KN.

The entire output of the pump passes through the orifice in accordance with Equation 4 which indicates

Fig. 4 — Pressures within the clutch for the three different conditions assumed in Fig. 3

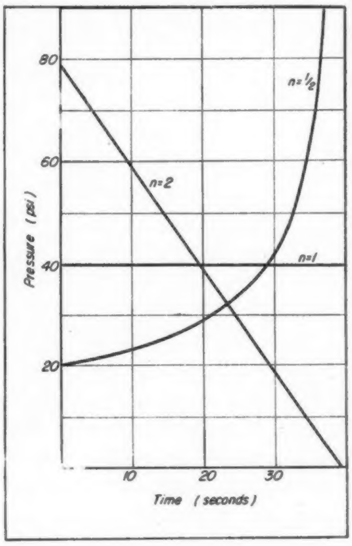
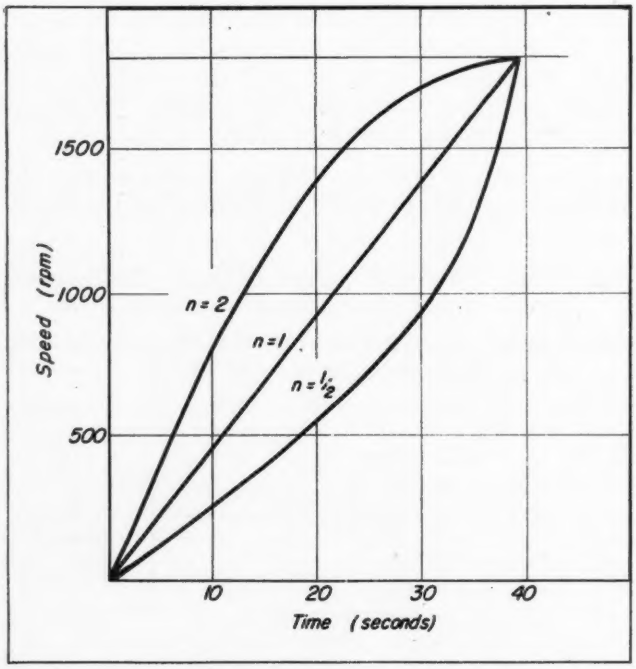


Fig. 3—Speed-time relationship during clutch engagement for three different designs. Low value of n denotes large initial orifice area and rapid rate of closing



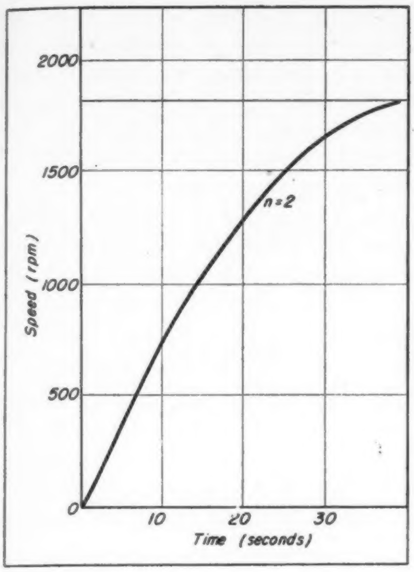
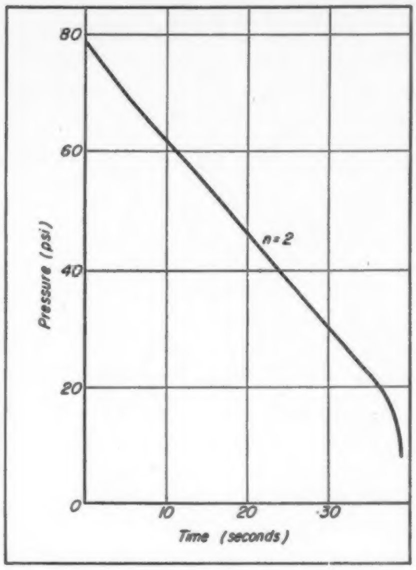


Fig. 5 — Left—Speed-time relationship with frictional torque considered. Curve lies slightly under the corresponding curve in Fig. 3

Fig. 6—Right—Pressure within clutch during engagement, friction being considered. Curve lies above the corresponding curve in Fig. 4



that the rate of flow is directly proportional to the pressure difference and the area of the orifice, which is reasonably accurate with liquids whose viscosity lies in the range of that of oils commonly used in this type of device.

If the orifice control mechanism effects a decrease in area, it is apparent from a consideration of the relationship of Equation 4 when written thus

$$P=\frac{Q}{CA} \qquad (5)$$

that P will increase as the area A is decreased.

Now the torque T exerted on the pump to drive it must be equal to the torque exerted on the load shaft if the mass and acceleration of the pump itself are neglected. It follows then that the torque applied to the load shaft is related to the area of the orifice and rate of delivery as follows, from Equations 3 and 5:

$$T=\frac{QD}{CA} \qquad (6)$$

This indicates that the torque increases directly with the delivery Q and inversely with the orifice area A.

An expression for torque in terms of speed may be obtained by combining Equations 1 and 5:

$$P=\frac{Q}{CA}=\frac{DN-C_sP}{CA}$$

or

$$P\left(1+\frac{C_s}{CA}\right)=\frac{DN}{CA}$$

and

$$P=\frac{DN}{CA\left(1+\frac{C_s}{CA}\right)}$$

or

$$P=\frac{DN}{CA+C_s} \qquad (7)$$

which, when substituted in Equation 3 yields:

$$T=\frac{D^2N}{CA+C_s} \qquad (8)$$

The equation of motion of the load may now be written in significant form. First, the general equation for the angular acceleration of a shaft is

$$T=J\,\alpha \qquad (9)$$

where α is the angular acceleration and the other terms are as before. Using the expression for the torque obtained previously and considering the friction torque,

$$\frac{D^2N}{CA+C_s}-T_f=J\frac{dN_l}{dt} \qquad (10)$$

where the angular acceleration α is expressed as dN_l/dt. However, N has been defined by:

$$N=N_i-N_l$$

hence, if the input speed N_i is constant,

$$\frac{dN}{dt}=\frac{-dN_l}{dt}$$

Equation 10 may now be written:

$$J\frac{dN}{dt}+\frac{D^2N}{CA+C_s}-T_f=0 \qquad (11)$$

If the orifice is closed in accordance with the expression:

$$A=A_o-C_ot \qquad (12)$$

it is possible to write

$$J\frac{dN}{dt}+\frac{D^2N}{C(A_o-C_ot)+C_s}-T_f=0 \qquad (13)$$

Solution of this equation yields the expression

$$\frac{N}{N_i}=\left[1-\frac{CC_ot}{CA_o+C_s}\right]\times$$

$$\left\{-\frac{T_fA_o}{N_iJC_o}\left(\frac{1}{1-n}\right)\left[\left(\frac{1-CC_ot}{CA_o+C_s}\right)^{1-n}-1\right]+1\right\} \qquad (14)$$

where

$$n=\frac{-D^2}{JC_oC}$$

Equation 14 may be used to obtain an expression

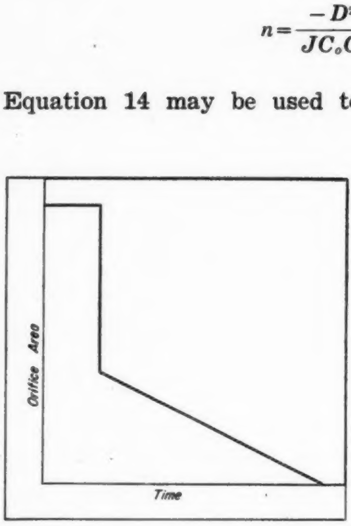


Fig. 7 — Ideal relationship between variable-orifice area and time for smooth clutch engagement

for the load speed N_l

$$\frac{N_l}{N_i}=1-\left[1-\frac{C_ot}{A_o}\right]^n\times$$

$$\left\{-\frac{T_fA_o}{N_iJC_o}\left(\frac{1}{1-n}\right)\left[\left(1-\frac{C_ot}{A_o}\right)^{1-n}-1\right]+1\right\} \qquad (15)$$

If the frictional load is negligible in magnitude and may therefore be neglected, the expression for the differential speed N reduces to

$$\frac{N}{N_i}=\left[1-\left(\frac{C_oC}{CA_o+C_s}\right)t\right]^n \qquad (16)$$

The load speed N_l may then be expressed in the following manner:

$$\frac{N_l}{N_i}=1-\left[1-\frac{CC_o}{CA_o+C_s}t\right]^n \qquad (17)$$

While neglect of the frictional torque represents an oversimplification, it does provide a useful means for studying certain physical characteristics of the phenomenon under consideration. It is immediately apparent that, as the clutch is engaged, the manner in which the load speed approaches the input speed depends upon the magnitudes of C_o, A_o, and n. For example, if the exponent n is equal to one, the ratio N_l/N_i varies linearly with time, until its value becomes equal to one. At that time the valve is completely closed and Equation 17 no longer properly describes the motion. However, this motion of constant acceleration would result in overshooting and oscillation after closure of the valve.

If the value of the exponent n is less than unity, the situation is worse since the ratio N_l/N_i increases at an increasingly rapid rate showing a theoretically infinite acceleration at the instant of orifice closure. This may be determined analytically by differentiating the expression for N_l (Equation 17) once with respect to time, to obtain the acceleration:

$$\frac{dN}{dt}=-n\left[1-\frac{CC_o}{CA_o+C_s}t\right]^{n-1} \qquad (18)$$

If the value of n is unity the exponent $(n-1)$ is zero and the quantity in the brackets raised to the power zero is one, thus yielding a constant acceleration equal to $-n$. If the value of n is less than one, the exponent $(n-1)$ is negative. When the orifice is closed $1-(C_o/A_o)\ t = 0$, hence the acceleration is given by the expression $(-n/0)^{1-n}$ which is infinity.

If the value of the exponent is greater than one, the acceleration of the load is initially great but decreases to zero at the instant of orifice closure since the quantity within the bracket is zero, and zero to any positive power is zero. This type of acceleration is to be desired and may be obtained by proper design.

The three types of load acceleration will be illustrated by simple examples and the operational characteristics of the clutch discussed in greater detail following these examples.

CALCULATED PERFORMANCE: Consider a hydraulic clutch of the positive-displacement type connecting a drive shaft to a purely inertial load. The following constants will serve to describe the system: Pump displacement D = 0.04 gal per revolution = 8.48×10^{-4} cu ft per radian; slip coefficient C_s = 0.002 gal per min × in.^{2} per lb = 3.08×10^{-8} ft^5 per lb-sec; moment of inertia of load J = 1 slug-ft^2; driving shaft speed N_i = 1800 rpm = 188 rad per sec; orifice coefficient C = 1 ft^3 per lb-sec.

The initial orifice area A_o and rate of orifice closure C_o are assumed different in three representative cases and have the following values:

A_o (ft^2)	5.6×10^{-6}	2.8×10^{-5}	1.4×10^{-4}
C_o (ft^2 per sec)	144×10^{-9}	72×10^{-8}	36×10^{-5}

With these values the time of closure for the orifice is 39.2 sec in each case, and the values of n are respectively 1/2, 1 and 2. Using the foregoing values of the constants the value of N_l was calculated for each type of operation by means of Equation 17, the results being plotted in *Fig.* 3.

As was indicated in the general discussion of Equation 17, the acceleration at the end of the clutch operation is excessive in two cases but is zero in the one case. The acceleration in each case is of course the slope of the curves shown in *Fig.* 3. It may be

demonstrated that, in the case of $n = 1/2$ and $n = 1$ in a practical situation, the angular speed will continue to increase after reaching 1800 rpm due to the fact that, while this system has been assumed to consist of rigid shafts, a real system would have elastic shafts. To assist in this demonstration the pressures within the clutch have been calculated and are shown in *Fig.* 4. Under the assumptions made thus far the torque exerted will be directly proportional to the pressure and in turn the acceleration is proportional to the pressure.

In the case of $n = 2$ the pressure, acceleration and torque are zero at the time of valve closure. It follows, therefore, that at that instant there is no strain in the shaft and there will therefore be no oscillation set up.

In the case of $n = 1$ the pressure, torque, and acceleration all remain constant during the period of orifice closure. Consequently at the instant of complete closure there is a torsional strain in the shaft proportional to the torque. When the load speed equals the input speed, the clutch pressure suddenly drops to zero and the elastic elements will oscillate causing a variation in load speed about the input speed as a mean. This situation is analogous to that which would obtain if the shaft were held fixed and the mass given an angular displacement and released. It would oscillate about the position of no displacement until the motion was damped out.

The situation for $n = \frac{1}{2}$ is similar, differing only in the magnitude of the torque and strain at the instant of complete closure. While the pressure is theoretically infinite, it would actually have some large value dictated by the physical characteristics of the system.

In *Figs.* 5 and 6 the effect of a constant frictional torque as a part of the load is considered. The frictional torque is assumed to have a value of approximately 1 lb-ft requiring a pressure of 8 psi to maintain constant speed. With a value of $n = 2$ the speed-time and pressure-time curves are as shown. The operation would still be satisfactory since the load speed approaches the input speed smoothly.

Practical Operation: In order to obtain the smooth operation which has been suggested as a definite possibility, further consideration must be given to the control of the orifice area. If the clutch connects a constant-speed driving motor to the load and it is desired to keep the load at rest while the driving motor is rotating, i.e., provide for disengagement of the clutch, it is necessary to open the orifice to an area sufficiently large to prevent the building up of the pressure to the magnitude required to start the load. In the ideal case of no friction torque, it is impossible to disengage the clutch. However, in all practical cases there will be some frictional drag.

In *Figs.* 7, 8, and 9 are illustrated the orifice area, the pressure and the load speed as a function of time in a practical case. Initially the orifice area is very large, *Fig.* 7, permitting the building up of a very slight pressure which is insufficient to overcome the frictional drag. When it is desired to pick up the load, the orifice area is decreased to a fraction of its original value. Since there is then a great difference between load and input speed, the pump delivers a large quantity of liquid, building up a large pressure to force it through the small orifice. As the load is accelerated, the pump speed decreases. This causes a reduction in pressure even though the orifice area also decreases, provided the pump characteristics and rate of orifice closure yield a value of greater than one. The resulting speed-time curve, *Fig.* 9, shows the desired characteristics. After orifice closure the pressure, *Fig.* 8, remains constant at a value sufficient to overcome the frictional drag and maintain constant load speed at a value slightly less than input speed. The difference in speed represents the speed necessary to pump the slip at the given pressure.

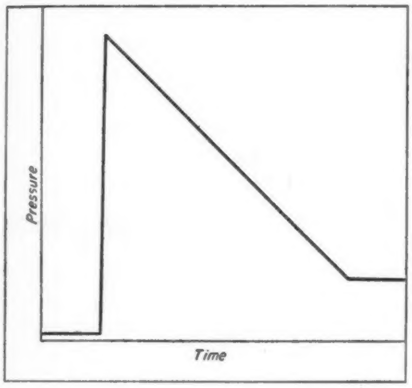


Fig. 8 — Left — Pressure within the clutch during engagement, using the orifice area variation shown in Fig. 7

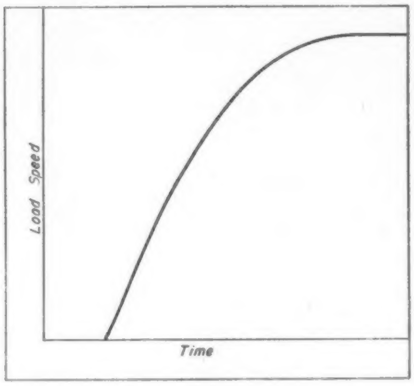


Fig. 9—Right—Speed-time relationship during clutch engagement, with orifice area variation shown in Fig. 7

It should be noted that the values for the constants of the clutch used in the problem were purely for illustrative purposes and are not intended to imply a sluggish clutch operation requiring a period as long as 39 sec for closure. The pressures built up were nominal for this type of service and a pressure of 1000 to 2000 psi would be in order, yielding an orifice closure time of the order of 1 sec.

Conclusions: The elementary analysis of the mechanics of the clutch indicates the possibility of suitable performance characteristics which are governed by orifice area and rate and manner of closure. Further analysis is in order to establish more completely the criteria for good clutch performance. Experimental analysis should accompany the mathematical to evaluate the theory.

Model Ansi

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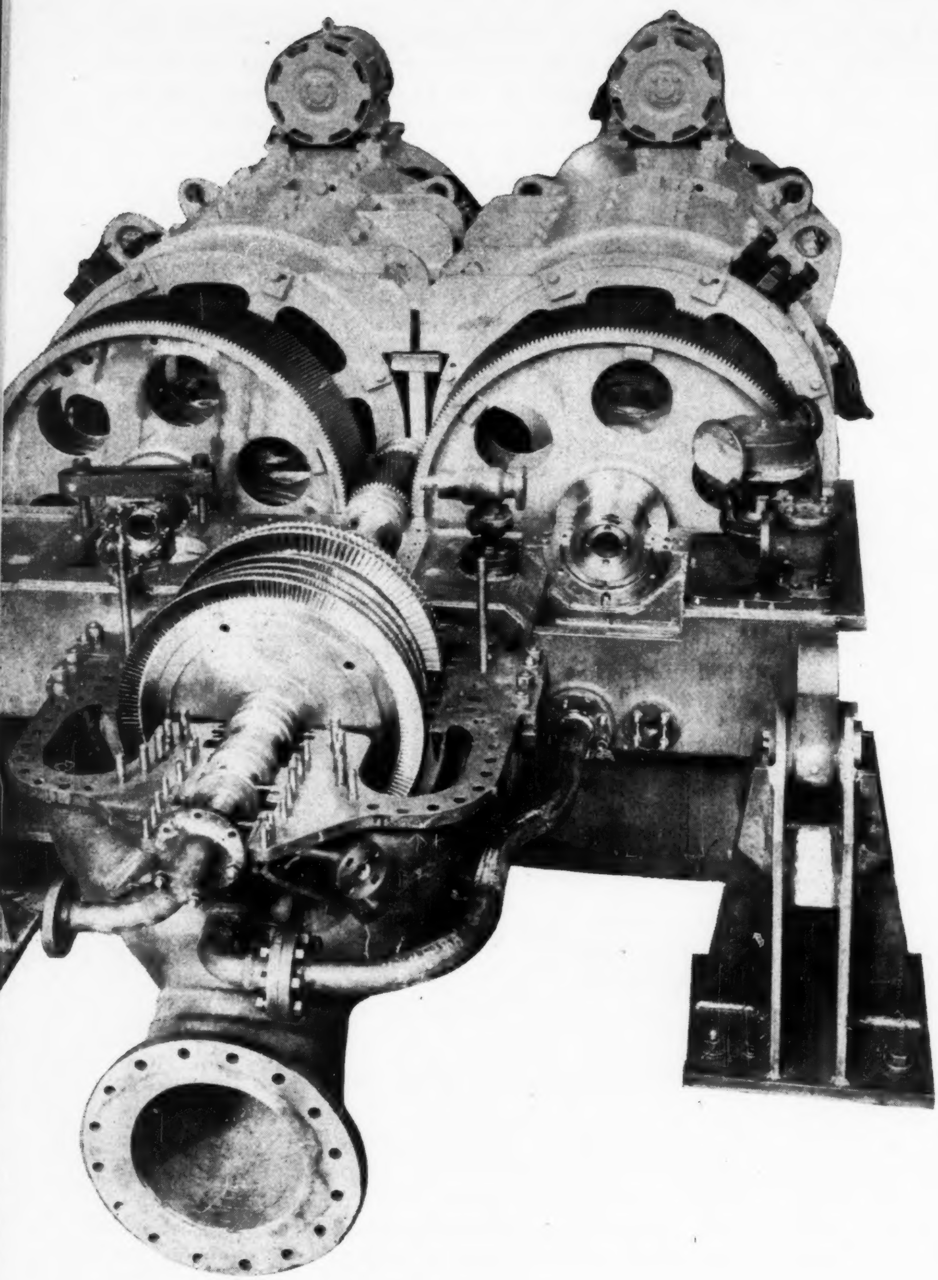


Fig. 1—Deflections in housing for this turbine-electric unit were analyzed with aid of the plastic model shown in Figs. 2 and 3. Unit is a single 6000-hp, 6000-rpm steam turbine driving two double-armature generators through a single reduction-gear to each generator

DESIGN of a turbine-electric unit for a locomotive presented many problems including that of insuring satisfactory alignment under all operating conditions of the rotating elements: Turbine rotor, pinion, and two mating gears solidly coupled to the generator armatures as shown in *Fig.* 1. It was found impractical to design a locomotive frame of sufficient rigidity to maintain satisfactory alignment of rotating parts. Therefore, an internally rigid, self-contained power unit was designed to be supported on the locomotive frame at only three points: Two trunnions at the forward corners and a flexible-rubber support pad between the generators.

The complexity of the structure made theoretical deflection analysis of sufficient validity most difficult. A ¼-scale plastic model, *Fig.* 2 made of Lumarith, having a modulus of 30 x 10^4 or about 1/100 that of steel, was then constructed which permitted the effects of the various loads to be determined conveniently. Deflection of the model and actual structure are related as follows:

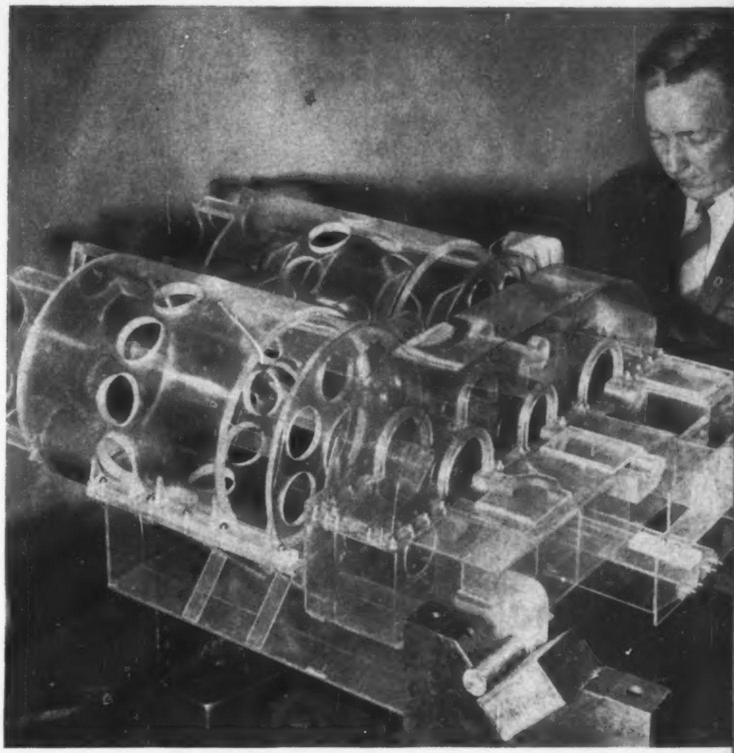
$$\frac{d_s}{d_m} = \frac{F_s}{F_m} \times \frac{E_m}{E_s} \times \text{model scale}$$

where: d = deflection, F = force, and E = modulus of elasticity. Subscript s is for the structure and m is for the model. Loads of 1/400 of the actual, therefore, caused model deflections theoretically equal to those of the real structure.

In mounting the model for test care was taken to duplicate the support restraint in the locomotive. At the forward end the model was supported on rigidly held mandrels duplicating the two trunnion supports. Rollers beneath the generator support pad allowed horizontal motion similar to that permitted by the actual rubber pad.

Loads to be applied included full-load torque, gravity loads of such parts as turbine, gear, and generator, and inertia loads. These inertia loads are developed when starting, stopping or rounding curves, as well as under conditions of normal pitch and roll in high-speed operation on straight track. They are, of course, variable in magnitude and direction. Maximum expected values are of the order of 2g, but are

nsis Aids Designer



normally $\frac{1}{4}$ to $\frac{1}{2}g$. These loads were simulated on the model by concentrated loads at the center of gravity of such major components as generator, turbine, gear and bedplate. Loads were developed by applying scale weights directly or over pulleys for directions other than down, *Fig.* 3.

It was found that the internal torque transmission resulted in negligible deflections and changes in alignment. Vertical inertia loads also resulted in practically no change in alignment although there was considerable deflection. Lateral inertia loads, as expected, caused a twisting of the structure due to the low center of twist of the section betwen the gear housing proper and the generator stators. The resulting deflection and misalignment of bearings on the generator and gear shafts and pinion was of sufficient magnitude that further investigation with the objective of increasing the rigidity was desirable. After making similar tests with numerous alterations, it was found that the tie plates between the generator stators and the gear cover were most effective, reducing the deflection and misalignment by about one-half for all loading conditions, with negligible weight addition.

A practical check of the model test data was later obtained by comparing the results of similar loading conditions on the model and upon the actual unit. The average difference was plus or minus 15 per cent. The usefulness of a model in solving problems of this type and contributing to design improvements has again been amply demonstrated. Probably no other approach could result in such a complete investigation and give results doubling the stiffness of such a complex structure with so little addition of weight.

Fig. 2—Top Right—One-quarter size scale model of unit supported similarly to actual unit

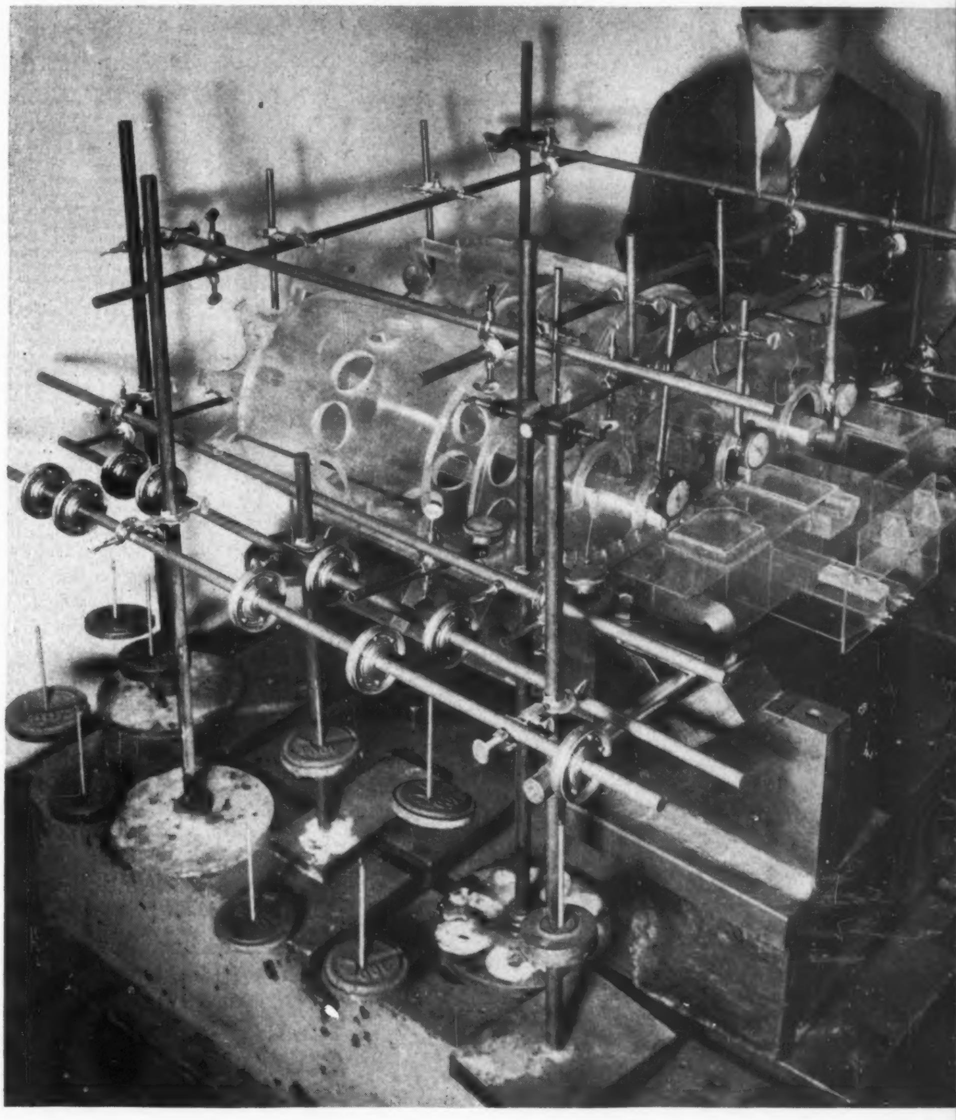


Fig. 3—Right—Plastic model loaded to simulate actual operating conditions in order to determine deflections

Adhesives...

Fig. 1—Many applications of adhesives are used in the construction of a power transformer. Taped joints are coated with adhesives, spacers are cemented pressboard, and support tubes and coil cylinders are impregnated paper

IN DESIGNING equipment for the electrical industry, the problems encountered are mechanical as well as electrical, with the former frequently as important as the latter. The adhesives used in the construction of such equipment frequently must retain their mechanical and electrical properties for long periods of time at elevated temperatures.

This means that an adhesive film may have to possess good dielectric strength, low power factor, and a high level of arc resistance. In certain applications, the adhesive must be essentially nonhygroscopic in order that electrical properties will not vary appreciably with changes in the relative humidity of the surrounding air. Typical characteristics and applications of some of the more important adhesives used for electrical equipment are listed in the accompanying tabulation.

For use with liquid-filled equipment, the adhesive must be of such a nature that it is not attacked by the insulating oil or chlorinated aromatic compound, nor must it have a deleterious effect on the insulating qualities of the liquid. Transformers, motors, generators, and other electrical apparatus are generally built to be operated with a 55 C rise, or at 80 C. Heavy units of this type, *Fig.* 1, are expected to have a useful life of at least twenty years. This requires, then, that the adhesives withstand twenty years ageing at 80 C without serious deterioration of bonding qualities or of electrical properties.

...ir properties and use in electrical equipment

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Frequently, temperatures higher than 80 C are obtained in class A insulated apparatus, since the maximum "hot spot" temperature may be allowed to reach 105 C. These relatively high temperatures for long periods of time impose ageing conditions which do not permit use of many adhesives commonly used in other industries.

Perhaps the first adhesive used in the electrical field was shellac. When used as an adhesive, it is applied, for instance, to two pieces of paper which are pressed together until the solvent has evaporated to the point that the viscosity is raised high enough for the shellac to act as a bonding agent. When the solvent has evaporated completely, a hard, rather brittle resinous film will remain which is a very strong adhesive.

In this condition, however, shellac is thermoplastic and will not retain much cohesive strength above 50 C. Consequently, it should not be subjected to mechanical stresses above such a temperature. The chemistry of shellac constituents is such that continued heating at elevated temperatures (125 to 150 C) will cause further polymerization and thus raise the softening point considerably. Cured shellac bonds can then be operated satisfactorily under some mechanical stress at 80 C. Shellac has so many desirable electrical and adhesive properties that large quantities are still used in the electrical industry despite the advent of many inexpensive and capable synthetics.

Other natural materials such as copals, wood pitches, asphaltics, and oleoresinous compositions are used extensively. They are applied as binders to cement cotton, paper, and other insulating materials and also used in hot melt or solution form as impregnating varnishes. Even today, large quantities of asphaltics and pitches are used as potting compounds to fill voids and cement electrical parts into containers. Impregnating and bonding varnishes comprised of blown asphalts and drying oils are presently used in large quantities for making a moisture resistant bond and coating

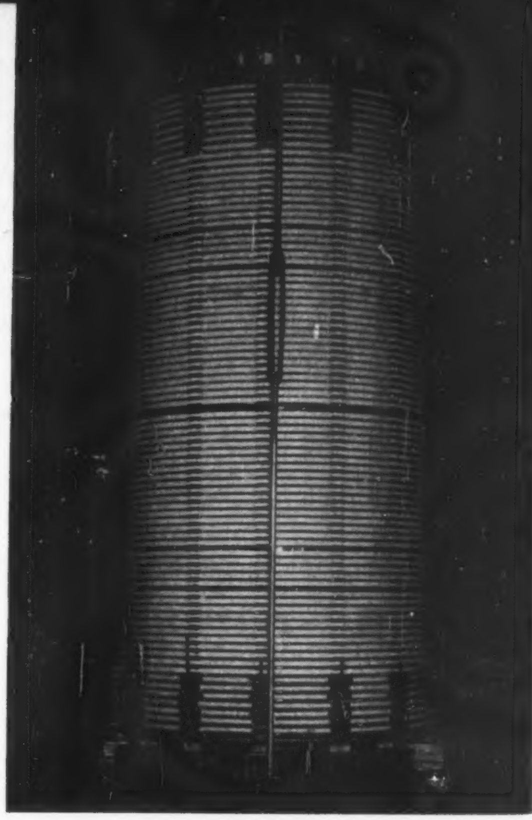


Fig. 2—Top Right—Varnish impregnation of this coil makes a permanently bonded unit of insulated wire, supports and spacers

Fig. 3—Right—Laminated phenolic-resin insulating collar, built of segments of pressboard. The many glue lines indicate the large number of segments used to make the insulation part

on rotating electrical equipment.

Naturally occurring adhesive materials, such as starch, dextrine, animal glues, gum arabic, silicates, etc., have been used for cementing cellulosic materials, such as paper and cotton. They are still used for splicing purposes, for sizing, and for bonding such materials. Such adhesives contain water as the solvent and can be made to develop a quick tack with porous materials like paper, which rapidly absorb the water from the adhesive.

Disadvantages of these materials, such as variation in quality, brittleness, or thermoplasticity has limited their use and caused their replacement in many applications by superior synthetic resins.

Bond Assemblies Into Unit

Varnishes are frequently employed for their adhesive properties. Completed coil assemblies, properly clamped, are impregnated with varnish, *Fig.* 2. After baking and setting of the varnish, sufficient mechanical strength is developed through adhesive action of the varnish to permit removal of clamps. The varnish film adheres tenaciously to adjacent turns and layers of insulated wire, to wooden supports, pressboard spacers, and laminated parts so that the coil structure is permanently bonded into a solid unit. In such applications the primary function of the varnish is as an adhesive.

Frequently an adhesive which has rapid drying properties and an affinity for many surfaces is needed for applications such as fastening the ends of cloth tapes, cementing spacer assemblies, holding paper insulation and varnished cloths. These materials may be fastened to themselves or to wood, copper, steel, etc. Adhesive compositions which have been widely used for these applications are modified nitrocellulose adhesives similar to air drying Glyptals, or similar cements. Typical use is shown in *Fig.* 1 where the adhesive is being brushed on a taped lead connection of a power transformer.

In *Fig.* 1 it is also interesting to note the adhesive applications represented. All the taped joints are coated with adhesive to keep insulation in place. The pressboard collar-spacer assemblies are cemented together, the lead support tubes are wound from paper and adhesive, the cylinders on which the coils are wound (not visible) are paper and adhesive, the spacing strips betwen the cylinders and the coils are pressboard laminated with adhesive, and the complete coil structures have been varnish treated to provide mechanical anchorage throughout. Similar to bonded parts used in this transformer are the laminated insulating collar, pressboard spacer assemblies and insulating cylinders, *Figs.* 3, 4 and 5.

Adhesive Tapes Replace String

A large market and use for pressure-sensitive adhesive tapes has developed in the electrical industry. The success of these tapes has resulted largely from the ability of the supplier to build into the tape the required electrical characteristics. Pressure-sensitive tapes are available with various backing materials, such as papers, acetate films, vinyl resin films, cotton, rayon, and glass cloths.

These tapes find their major use in assembly operations, particularly where there is no permanent dependence upon the adhesive bond. An ideal application is in the assembly of motor coils. In this case, it is desirable to hold in shape coils of wire previously wound on a machine. The old practice was to tie each coil with a string. This costly operation has been practically eliminated by the use of a pressure-sensitive tape, which is quickly applied to the coils while they are still on the winding frame.

Typical Properties of Some Adhesives

Adhesive	Type	Flexibility (ASTM D-522)	Oil Resistance at 100C	Moisture Vapor Transmission (g/hr/cm/cm²/mm Hg)	Dielectric Strength* (volts/mil)	Dielectric Constant	Power Factor 60 cycle (%)	Arc Resistance (Sec)	Steel to Steel bond (TS in psi)	Heat Resistance	Typical Applications
Nitrocellulose & alkyd resin	air-dry	<5%	excellent	$10\times10^{-}$	1500		4	70		fair, becomes brittle	Rubbers Bonding cellulosics Good bonds to metal
Vinyl acetate	air-dry	excellent	good		1500		4			fair	Bonding cellulosics
Cellulose acetate	air-dry	brittle	excellent		2500					good	Bonding cellulosics
Dextrin	air-dry	brittle	excellent							good	Bonding cellulosics
Phenol-butyral	air-dry & bake	>30%	excellent	3.5×10^{-8}	2000	3	5	125	2000-6000	excellent	Metal to metal bonding. Bonds to cellulosics
Liquid ester resin	Heat cure	brittle	excellent	4.3×10^{-8}	2000	5.4	1.3		500	excellent	Glass cloth laminates
Resorcinol Formaldehyde	room cure	brittle	good		200	20	18	15	10	good	Bonds cellulosics and some plastics
Alkyd	bake	>30%	excellent		2500	2.7	20	84	250	excellent	Bonds cellulosics
Shellac	air-dry or bake	brittle	excellent		2000						Bonds cellulosics, metals, plastics, glass, etc.
Modified neoprene	air-dry	>30%	poor		700	5	7	125	200-700	excellent	Bonds metals, rubbers, cellulosics, plastics

*Dielectric Strength is based on 5-mil, 60-cycle, short-time test.

These coils are subsequently assembled into a rotor or stator, which is varnish treated to provide a permanent adhesive bond.

Much electrical equipment is sealed in containers to prevent the entrance of moisture and to protect the apparatus from external factors. The container for liquid-filled apparatus is frequently sealed with a composition cork gasket. In such cases, it is necessary to have an adhesive that will bond cork to metal, porcelain or other materials. The adhesive must be moisture resistant, fit the assembly process, and not affect the insulating liquid.

Many electrical applications such as light bulbs, electron tubes, porcelain enamels, and insulating bushings require a glass to metal seal. This requires an intimate bond. Such a bond seldom involves the use of an adhesive as such. The glass to metal bond is dependent upon a primary chemical reaction between the two substrates, and it has been found important to match the coefficients of expansion of the substrates so that glass fracture does not occur on temperature change. It is helpful also to have the metal thin and pliable.

Insulation Must Be Flexible

Another interesting problem of adhesion is in the enameling of copper wire with an insulating coating. It is necessary in this case for the bond between the copper conductor and the thin insulating film to be sufficiently good so that the film will elongate with the copper when stretching occurs. Also, elongation and compression of the insulating film must take place when the wire is bent, otherwise the film will break loose from the wire and crack. In certain cases, the poor adhesion is evidenced by "tubing" of the film when the wire is stretched.

Within the past few years a group of new adhesives has been introduced which have tended to revolutionize the bonding materials by replacing the older types, to some extent, and more importantly, by opening a whole new panorama of adhesive effort. These materials are responsible for the wide use of metal to metal bonding, the sandwich constructions involving light inner cores laminated to metal, and the easy high strength bonding of rubber to metals and other substrates.

Adhesives of this type generally comprise a solution that can be applied by brush, spray, or spreader. After application, the solvent is completely removed by air drying or a moderate bake. Then the surfaces are pressed together and heated. During this cycle, the adhesive softens, and, as the baking continues, it begins to cure or set so that it becomes tough and strong even at elevated temperatures. The whole effect is produced by a series of chemical reactions.

A thermoplastic resin, such as Alvar, which by itself is flexible and a good adhesive, is combined in solution with a compatible thermosetting resin, such as a phenolic. When the adhesive solution is brush coated on a surface and the solvent evaporates, a hard film which contains the uncured thermosetting resin is left. When heat and pressure are applied the thermoplastic resin softens and its flow is aided by the plasticizing action of the uncured thermosetting resin. As the heating continues, the thermosetting resin begins to harden and become strong; when the cure is complete the adhesive consists of a strong thermosetting resin plasticized by a strong, flexible, thermoplastic resin.

The particular effect of the thermosetting resin is to increase rigidity and temperature resistance. There may or may not be a chemical reaction between the thermoplastic and thermosetting portions. Phenolic resins are known to contain incompletely reacted portions such as saligenin. These can react with an unsaturated material to form a chroman. Adhesives of this type are frequently used for bonding metal to metal. The following illustrates how such a bond may be made.

A lap joint, one inch long, was required between pieces of steel 1/16-inch thick, 6 inches wide, and 12 inches long. The surfaces to be bonded were carefully sanded on a 240 grit belt to expose bright clean metal, immediately washed with clean trichlorethylene and allowed to dry. As soon as this was completely dry, a dilute solution of adhesive was brushed

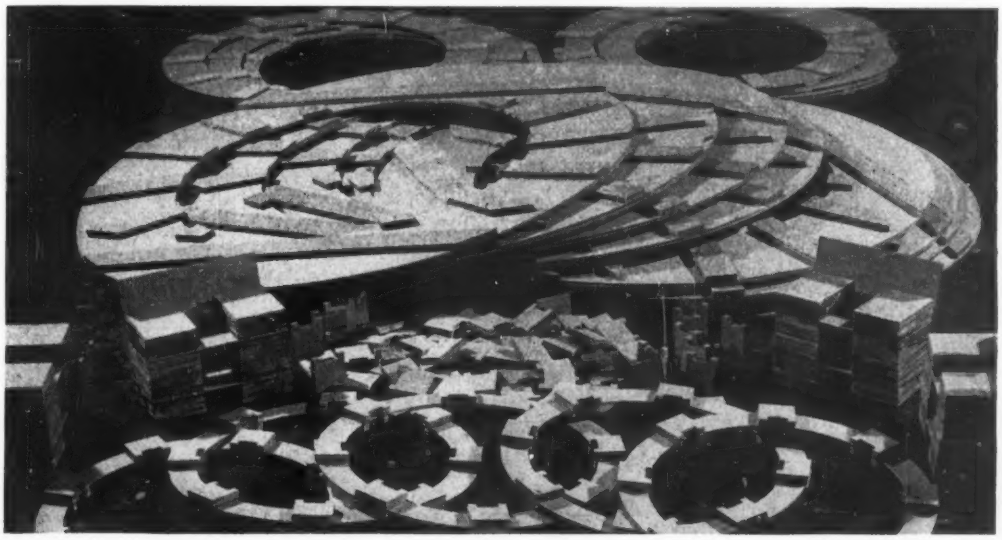


Fig. 4—Spacer assemblies of laminated pressboard bonded with phenol-butyral, showing some of the glued structures involved in power transformer insulation

on both surfaces to be joined and allowed to air dry thoroughly.

A sufficient number of coats was applied to build a film thickness on each piece of 0.002-inch. After the last coat was applied, the pieces were heated in an 80 C oven for one hour to remove any residual solvent. After cooling the samples were assembled into a jig and placed in a heated hydraulic press. A temperature of 150 C was maintained for 45 minutes at a pressure of 1000 psi on the glue line. After the heating cycle, the joint was cooled under pressure. It was completely effective.

Heat and pressure bonding adhesives were used extensively during the war for cementing metal lap joints in aircraft wings. Such joints were found to be stronger and more dependable than riveted ones. Also, these adhesives have been used in cementing thin metal sheets to plywood cores in sandwich constructions. These light weight and strong assemblies are of particular value in aircraft applications. Similar constructions, where the plywood might be replaced by plastic electrical insulating material, are of value in electrical applications.

Adhesives are frequently used to bond rubber to metal for providing mounts. By first treating the rubber surface with concentrated sulfuric acid for a few minutes, washing with water, drying, and flexing, the surface of the rubber is changed so that the adhesive can bond tenaciously. Properly prepared joints between rubber-like materials and metal are easily made strong enough so that severe testing will cause rubber failure rather than adhesive failure. This high-strength bonding of rubber to metal warrants further study.

Fig. 5—Heavy-wall, phenolic-resin impregnated insulating cylinders for use where high mechanical and high electrical stresses are involved in electrical equipment

Such adhesives can be used to prepare strong bonds beween and to metal (steel, iron, aluminum, copper); cellulose materials (paper, cotton cloth, wood); vitreous materials (glass, porcelain); plastic materials (thermoset resins, laminates) and rubbers (natural, neoprene, Buna N, Buna S).

Solvent activated adhesives depend upon the removal of solvent for their bonding properties. An example of this type might be a properly processed neoprene synthetic rubber dissolved in toluene or some such solvent. When the adhesive solution is brushed onto two surfaces and a portion of the solvent evaporates, a strong tack develops. This tack persists for a few minutes, and if the two surfaces are brought together during this period, they will adhere firmly.

Are Useful for Bonding

Such adhesives are useful in making bonds betwen synthetic rubber pieces, and between rubber and metal. To cite a specific example, the preparation of the scarfed joints in a synthetic-rubber gasket for a transformer tank involved the following: The cement selected was made of the same type rubber as the gasket. The rubber strips were scarfed in a proper device to prepare a piece as indicated in the shear-test method. The scarfed surfaces were cleaned with acetone and a thin even coat of the adhesive applied to each surface. When a strong, aggressive tack has developed the surfaces were pressed together and the joint was immediately strong. Its strength could be improved somewhat by aging at room temperature. Heating in a press at 130 to 150 C removes residual solvent, which reduces the plasticity of the adhesive and strengthens the joint.

Adhesives made from materials such as cellulose acetate, vinyl acetate solutions or emulsions, Alvars, ethyl cellulose, and shellac can also function by solvent evaporation. Such adhesives are used largely in the electrical industry for assembling insulating pieces which will not subsequently have stresses at temperatures over 50 C or in some cases 80 C. These adhesives may be activated by heat and pressure after the solvent has completely evaporated. They are strictly thermoplastic and so must be cooled before removal from the press or mold.

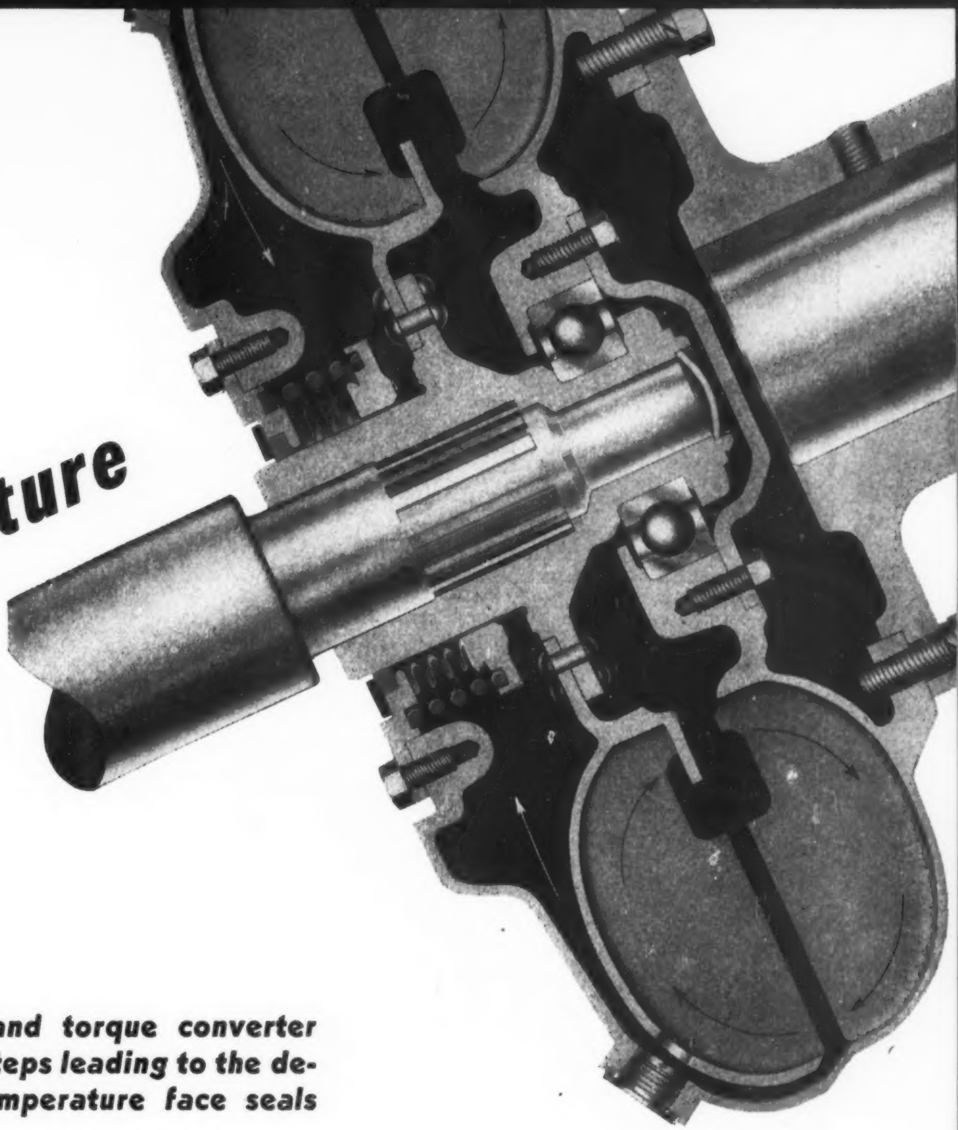
Among the cold-setting adhesives, a new type consists of an incompletely reacted resorcinol-type resin and a dry powder which contains catalysts and hardeners. Such adhesives are used by first mixing the powder in the adhesive and then applying. In a period of time depending upon circumstances, 2 to 24 hours, the adhesive sets at room temperature to a hard, rigid material.

As an example of such material, a joint was made between two pieces of flat plastic laminate, the laminate being made of phenolic resin and paper. The plastic sheets were scarfed in a suitable jig, and a coat of freshly mixed adhesive applied to one portion of the joint. The pieces were then fitted together under a moderate pressure and allowed to stand at room temperature. After four hours the clamps could be removed, and after forty-eight hours the joint was ready for severe testing. The forty-eight hour period can be reduced to a few hours by heating at 80 to 100 C to speed curing. Such a joint can be 80 per cent as strong as the laminate itself.

This process makes it possible to manufacture

(*Concluded on Page* 178)

High-Temperature Face Seals



How hydraulic coupling and torque converter seals were developed . . . steps leading to the design of successful high-temperature face seals

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EARLY torque converter and fluid coupling seals were based on construction principles which involved conventional metal-to-metal labyrinth designs. This type seal served satisfactorily in large marine and industrial applications, primarily because an enclosing housing could be used from which any oil loss could be pumped back into the system. Adaptation of these units to automotive applications, however, immediately made apparent the need for a positive seal to prevent oil loss. Obviously no enclosing oil housing could be used. Maximum simplicity of design was imperative and any oil loss would affect efficiency.

Work began, therefore, about ten years ago toward the perfection of a good fluid coupling and torque converter seal

Fig. 1—Top—Industrial hydraulic coupling showing free-floating seal ring which seals drop-tight for life of unit

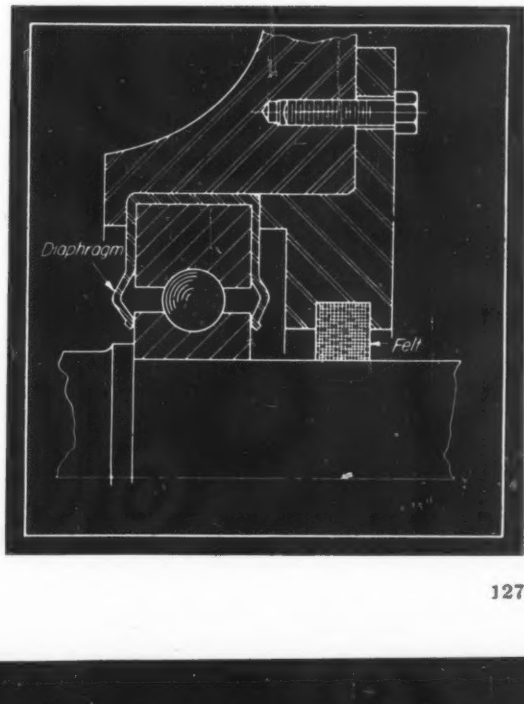


Fig. 2—Right—Simple diaphragm-type sealing ararngement was the first attempt to meet the coupling problem

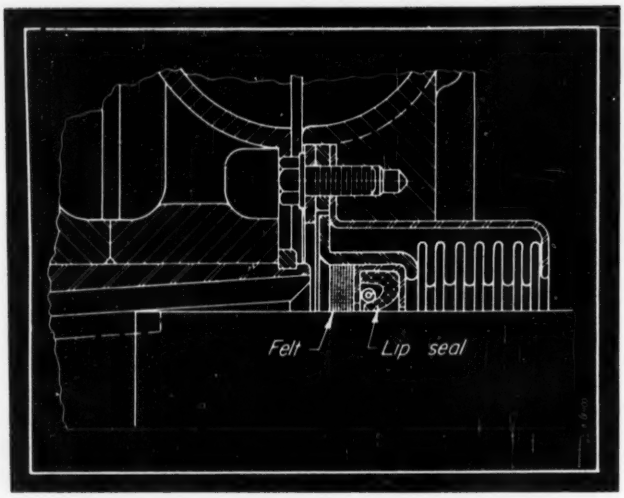
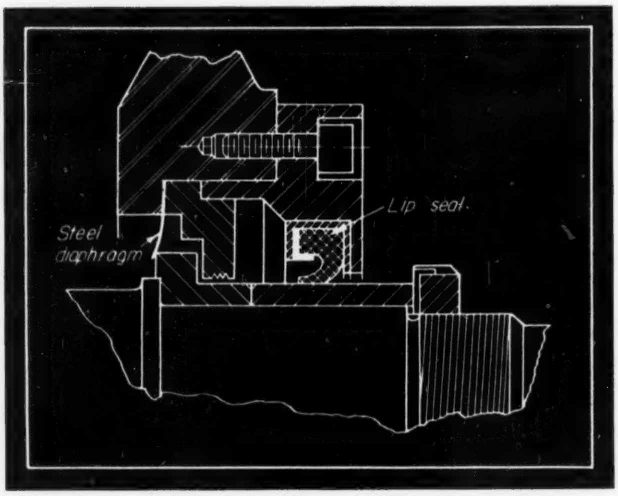


Fig. 3—Above —Lip-type standard seal, the second attempt, failed primarily from lack of self alignment

Fig. 4—Below—A variation of the lip seal involving a spring-steel diaphragm also failed to satisfy requirements



with the result that the units now manufactured seal drop-tight to prevent any leakage loss whatsoever, *Fig.* 1. It is interesting to see how these high-temperature seals were developed.

Fluid Coupling Seals

A fluid coupling is an excellent means of transmitting torque through a liquid with a minimum loss in efficiency. It can deliver the same torque at about the same or any lower speed by use of the slip feature in the coupling. The fluid generally used is SAE 10 oil and automotive applications in general can have a maximum slip of 100 per cent at about 35 to 45 per cent of engine speed. Industrial applications run up to 1800 rpm with a 200 per cent slip when the driving member is stalled and then reversed. The slip differential represents lost power that turns up as heat in the oil and temperatures ranging up to 550 F can and are realized with a corresponding pressure of 80 psi. The coupling, in operation with minimum slip (3 per cent), carries oil out to the working circuit to leave the oil seal area relatively dry.

Thus a fluid coupling seal is required to withstand speed, pressure and temperature and a degree of partially dry operation. To compensate for manufacturing limits and minor alignment variations, it must also have ample axial movement as well as lateral and angular flexibility. Added to this list is a pressure blowoff requisite, i.e., any abnormal pressure condition that might arise must vent through the seal. Finally, cost must be reasonable.

The first shaft seal, *Fig.* 2, was a simple metal diaphragm on each side of a ball bearing. Under noncritical operation these seals had a degree of success but would not meet the high-pressure conditions of automotive service. A high pressure in the coupling meant a higher pressure on the seal, and this made it fail at the very moment that it was needed the most. The next seal tried was the standard lip-type shaft seal, *Fig.* 3. This seal failed under both pressure and temperature. The lip would turn up and once leakage started, it continued. Other designs, *Figs.* 4 and 5, had similar limitations.

Reverting back to a face-type seal, the single and double rotating shaft seals were tried, *Figs.* 6 and 7. These are good seals for ordinary service, but the synthetic rubber elements could not stand up under the high temperatures encountered (550 F) with fluid couplings. In the search for a face-type seal

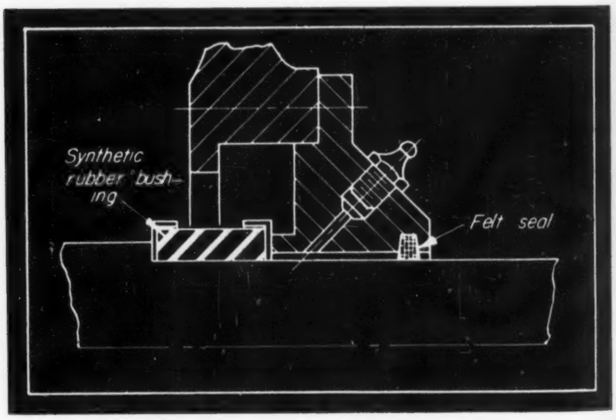
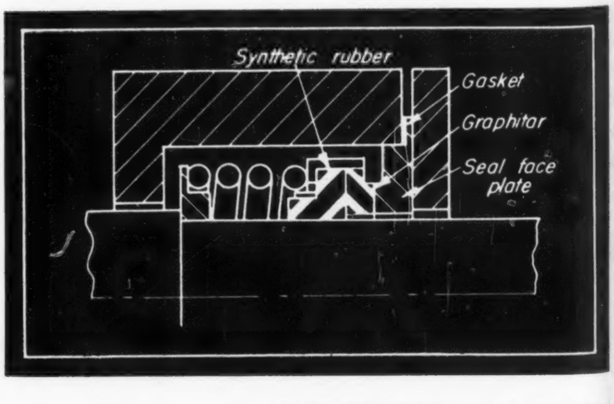


Fig. 5—Above—Seal employing cyanide-hardened and lapped steel washers on a synthetic-rubber bushing also failed to operate successfully

Fig. 6—Below—Single-type rotating shaft seal for ordinary service would not withstand high temperatures



without a temperature sensitive member, a single-ply unit metal bellows seal was tested, *Fig.* 7. A comparatively high degree of sealing resulted, but the semidry running requirement and the need for great axial flexibility (0.093-inch before wear) were two serious obstacles. The bronze nose piece of this seal, running on a lapped steel seat would not work without oil. Too, once locked into positon, this seal would not give a uniform seal pressure when balanced against the hydraulic pressure—at different shaft positions—and still allow a constant blowoff.

Finally, in checking the popular refrigerator seal, *Fig.* 9, containing the metal bellows with a lapped high-lead bronze nose piece running on a lapped, hardened-steel mating member, it was found that this combination had only a small percentage of seals that would not hold. After reviewing all conditions, it was hard to see why this unit would not work 100 per cent. Then, the semidry running condition became more apparent. It was seen that constant operation threw all the oil out to the tubular part of the coupling and, as this seal tended to dry out, it would gall. Subsequent stopping threw oil under pressure on the static seal, and those few that had worn enough were leakers.

Knowing that the metal-bellows seal was basically

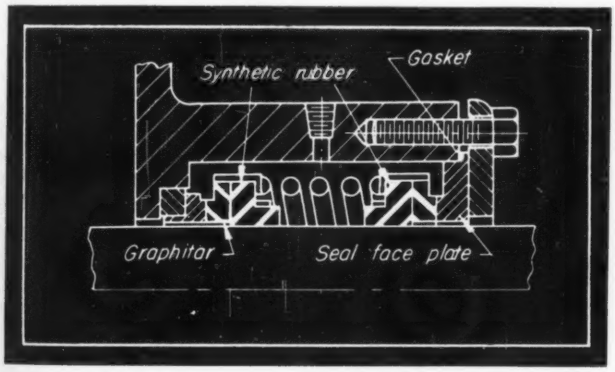
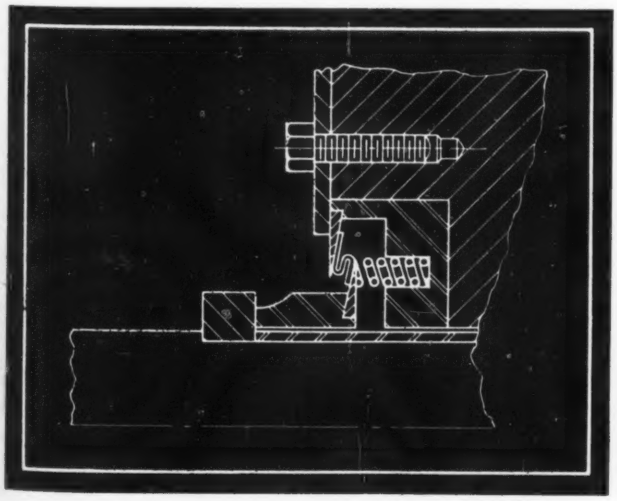


Fig. 7—Above—Double-type rotating shaft seal arrangement had same drawbacks as single seal

Fig. 8—Below—Single-ply metal-bellows type resulted in good sealing but lacked axial flexibility and self-lubrication characteristics which were necessary



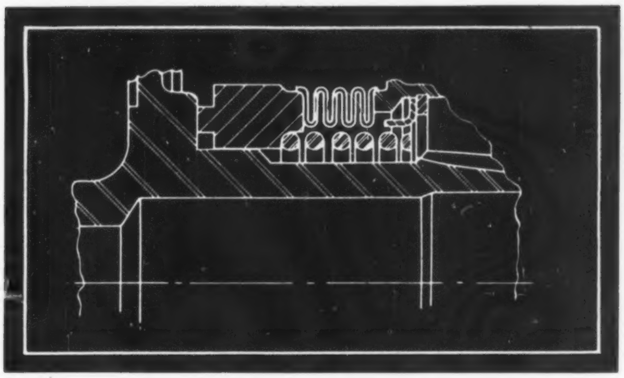
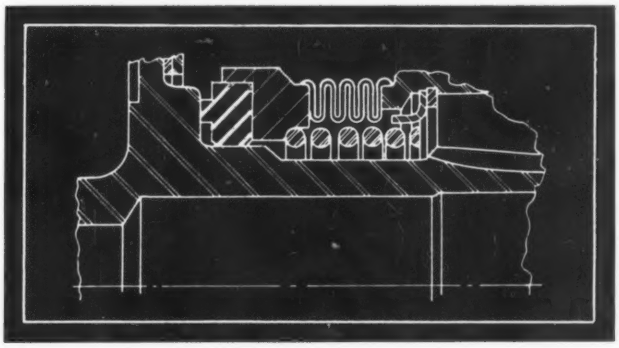


Fig. 9—Above—Bellows seal with a bronze nose, used in refrigeration work, failed to operate without lubricant

Fig. 10—Below—Carbon material substituted for bronze nose failed due to thermal expansion differences



sound, a search was made for a nose piece that could afford rubbing contact without galling. Glass and carbon materials were considered. Glass presented a production and assembly problem in that it could not be lapped easily and a satisfactory method of attaching it to the bellows was difficult to find. Nevertheless, glass was tested but found to warp out of flat and squeak. The variation in temperature coupled with the lack of surface lubrication at the seal seat were the primary reasons for the failure.

By this time the seriousness of this sealing problem was fully realized and, to work correctly, the nose-piece materials had to:

1. Be stable under temperature changes
2. Rub without galling
3. Have no wear or a low uniform rate of wear
4. Seal 100 per cent.

Carbon materials seemed to offer a good possibility. For the first try, a carbon material was brazed into a metal-bellows arrangement as a nose piece, *Fig.* 10. This seal gave encouraging results with consistent sealing at ordinary temperatures. Operating variations however showed that the high rate of expansion of the bronze and the low rate of expansion of the carbon developed unequal strains in the carbon nose piece. A lapped flat seal at room temperature might or might not go out of flat at 300 F, 400 F or 500 F and this too was objectionable. Therefore a further step was necessary to free the carbon.

The step from two lapped surfaces to four was a bad move from a cost standpoint but the free-floating design was made, *Fig.* 1. Lapped flat the free-

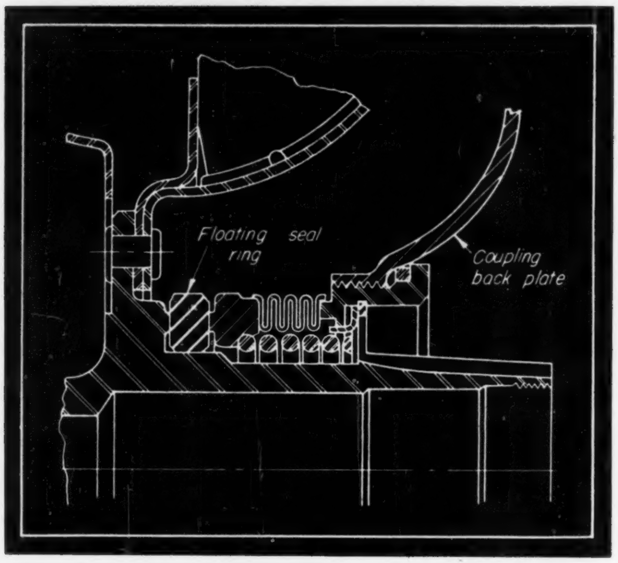


Fig. 11—Final seal design employs a carbon-graphite material of Rockwell C60 hardness

floating member would stay flat at all temperatures. A high percentage of consistently good seals resulted. The seal would take rubbing in the semidry condition without galling and the rate of wear proved to be extremely low. This seal design was adopted for production and used almost universally on the first fluid couplings made in large numbers.

Use of a free-floating carbon seal having two lapped surfaces on two wearing lands involved twelve grinding operations. This was expensive so the manufacturer investigated production of the seal from a powder metal alloy of some type. Innumerable tests were made in the laboratory to try and get consistent sealing. The final conclusion showed that a powder metal alloy seal ring was either too porous or too nonporous. If the seal was porous enough to lubricate, it would also leak through the pores. If it was nonporous enough to seal, it would score or gall. The basic material lacked the inherent lubrication necessary.

Specifications for flatness of the carbon member as well as the steel mating members of this seal proved to be one of the most interesting points in the development of the present-day design. The first seals were blued-in on a surface plate and had to show a 95 per cent contact. This was easy to do on the carbon and bellows sealing members but next to impossible to do on the hub surface. Seeking a better production checking method, the manufacturers found that the degree of flatness of a seal face could be measured by the rate of loss of vacuum applied between it and a lapped flat steel disk. This, however, did not provide a satisfactory means for controlling the flatness on the hub surface.

An optical flat with a monochromatic light was finally adopted and a careful study showed that a maximum out-of-flat condition of three light bands (35 millionths of an inch) on any one surface was all that could be allowed. Seals must seal without a "run-in" period and a maximum out-of-flat condition of over 0.00010-inch on two mating surfaces would cause a leak. Therefore, 0.00007-inch was set as the top limit for any two running faces.

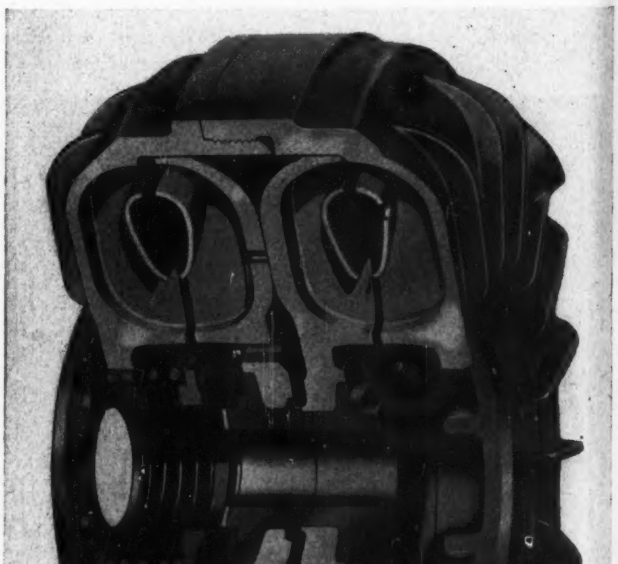
One carbon material of medium hardness had difficulty in meeting the close flatness specification as measured by the optical flat. Too, this design, *Fig.* 1, required the wearing surfaces on the carbon to be smaller than the mating steel surfaces, so a new design was made and Graphitar carbon-graphite material was adopted, *Fig.* 11. By using a Graphitar material with a hardness of about 60 Rockwell C, the seal lands that control the blowoff pressure could be transferred to the metal mating surfaces. This saved on lapping cost due to the smaller area of metal to be lapped. The seal was more stable as a material and could meet the three light band specification with less than a 1 per cent rejection.

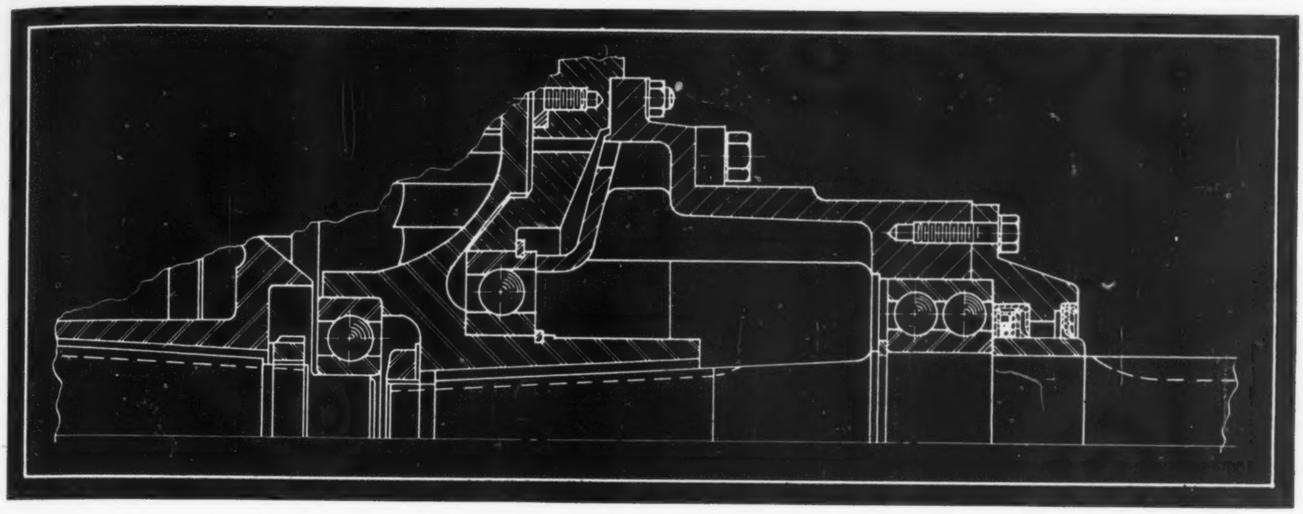
This seal design is used widely today, *Fig.* 11, and meets all seal requirements. Road tests show wear rates of less than 0.0001-inch total wear in over 100,000 miles of service. Its only drawback is the cost of the entire seal assembly as compared to the fluid coupling as a whole. To help correct the cost angle, one manufacturer of fluid couplings studied the seal cost breakdown and learned that lapping of the hub surface was one of the most expensive operations. Therefore, a new seal design was put into production, *Fig.* 12, using a synthetic rubber member to back up the carbon-graphite seal. This is permissible on this design as temperatures over 300 F are seldom experienced in industrial applications.

Torque Converter Seals

Kerosene or No. 1 fuel oil is usually used as the medium for transferring the torque in a torque converter. Five thousand gallons a minute can be handled at a speed of 3400 feet per minute at the seal diameter. This high velocity is amplified by the turbine blading when the pump is delivering its maxi-

Fig. 12—Somewhat lower in cost than the unit of Fig. 11, a seal requiring only one set of lapped faces employs a synthetic-rubber bushing behind the seal member





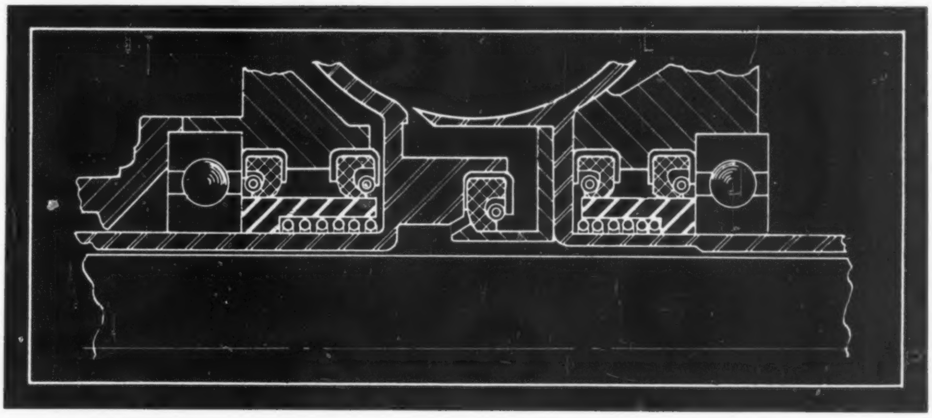
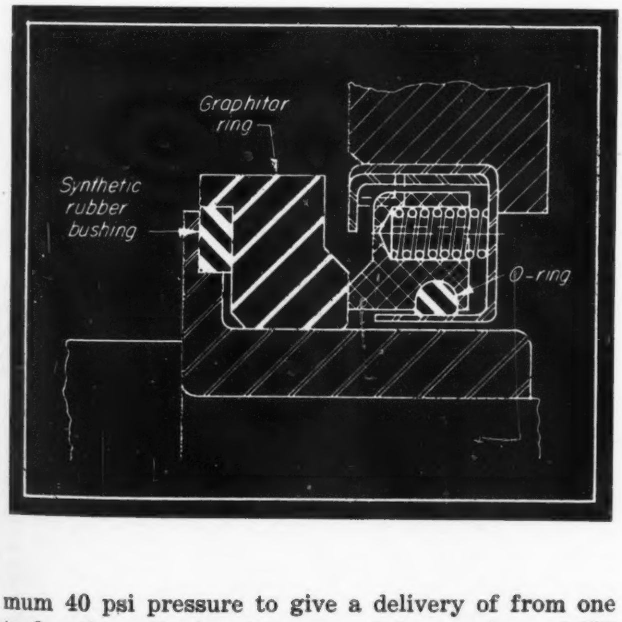


Fig. 13 — Above — Lip-type shaft seals proved as unsatisfactory on torque converters as on hydraulic couplings

Fig. 14—Right—Lip of shaft seals, placed in nonrotating contact with a keyed carbon sleeve, lacked sufficient self-alignment and flexibility

Fig. 15 — Below — Standard axial seal unit, used in conjunction with a carbon-graphite seal ring set in rubber, lacked axial adjustment



mum 40 psi pressure to give a delivery of from one to five times the input torque. A temperature of 275 F is developed in the stationary housing.

Manufacturing limits, coupled with pump and driven rotor movements due to heat variations and the high oil velocities make a maximum flexibility in a torque converter seal a necessity. Three seals are used; one between cover plate and pump, another between the pump and the driven rotor, and a third between the driven rotor and the stationary housing. These seals will not dry out but the fluid being handled is difficult to confine. Torque converter seal development has been as interesting as the fluid coupling seal development. It was independently tackled by three engineering departments of three separate companies and resulted in three designs.

Preliminary work soon proved that the lip-type seal, *Fig.* 13, was not much better than the metal-to-metal labyrinth seal for torque converter application. Therefore, one manufacturer took the lip seal off the rubbing surface and placed it on the outside diameter of a carbon sleeve, *Fig.* 14. The carbon sleeve, backed up by a spring, could then take the rubbing contact and the lip-type seal could seal on a stationary, stable member. A fair degree of sealing resulted; however, this seal did not have enough flexibility to stand up. Changes in alignment, as mentioned previously, turned up the lip of the seal and once leakage started, it would not stop.

Another manufacturer adopted a popular unit-enclosed seal using a Graphitar ring set in synthetic rubber for the sealing element. This sealed well but lacked the great amount of axial movement necessary, *Fig.* 15.

The third unit, a diaphragm-type seal is a design used on equipment today. It consists of a synthetic-rubber diaphragm attached to a carbon seal ring by means of a metal retaining ring, spun in place, *Fig.*

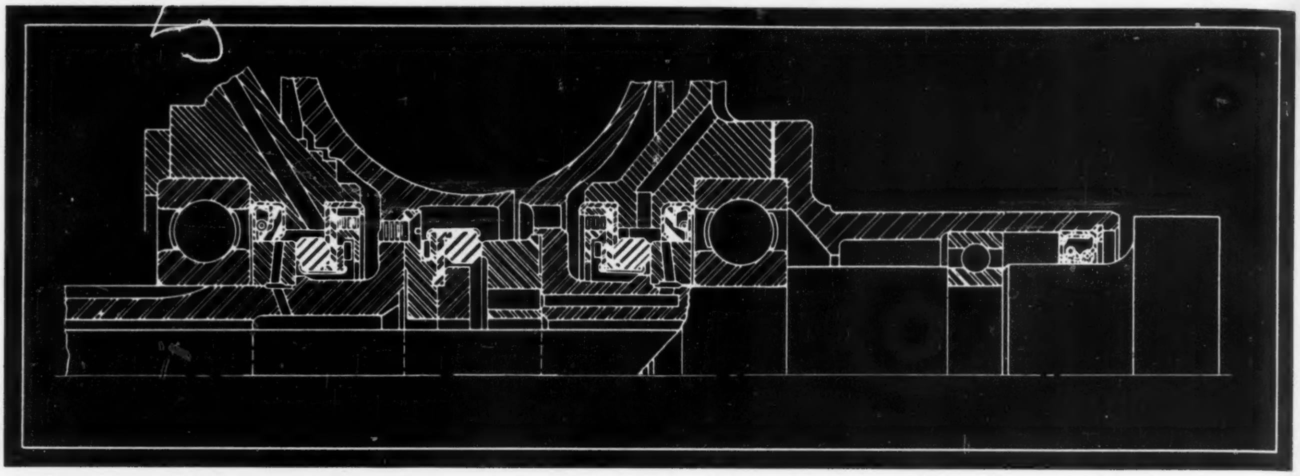


Fig. 16—Synthetic-rubber diaphragm attached to a carbon-graphite seal ring provided a highly successful unit

16. It is "run-in" on the mating surface.

When these latter two seal designs proved to have limitations, service reports on fluid couplings were closely studied and it was decided to adopt the metal-bellows seal with the free-floating ring, *Fig.* 1. Application of the fluid coupling seal to a converter was difficult due to the difference in the fluid used and the larger seal diameter required. It was much more difficult to lap the seal seat for the free-floating seal-ring mating surface so two separate methods of sealing off this face of the carbon-graphite ring without imparting its flatness were developed. One used a rather large synthetic-rubber ring on an outside-diameter shoulder of the ring, *Fig.* 17. The other solved the problem with a very thin gasket behind the seal. This second unit was made positive by a uniform pressure of a Belleville spring on the seal, *Fig.* 18. Each gave good results.

Ten years of development in making seal units for these hydraulic transmissions has provided the proof that a carbon-graphite material such as Graphitar has the best qualifications for the seal rubbing member. The material can be lapped flat to stay flat over a wide temperature range. It has the inherent lubrication necessary for semidry operation. It cannot melt, so it will not gall. These are the salient requirements of any seal material.

Material assistance of the following organizations in the preparation of this article is hereby acknowledged with much appreciation:

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General Motors Truck & Coach Div. G. M. Corp. .... Pontiac, Mich.

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Fig. 17—Converter seal design employs a synthetic rubber-mounted seal ring and bellows arrangement similar to that used with couplings

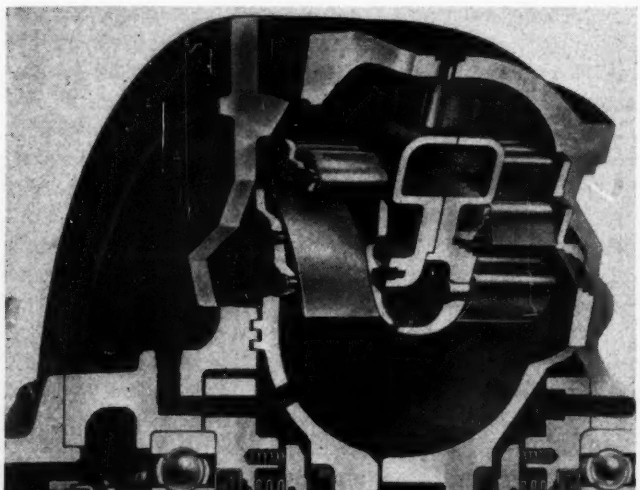


Fig. 18—Another version of the unit in Fig. 17 employs a gasket behind the seal ring with a Belleville spring to provide uniform loading pressure

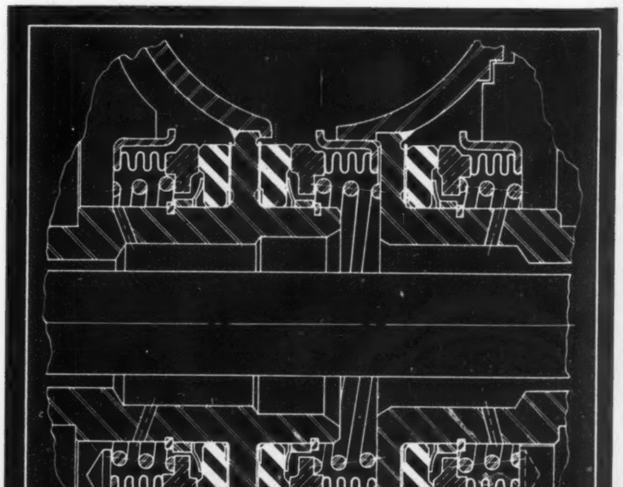
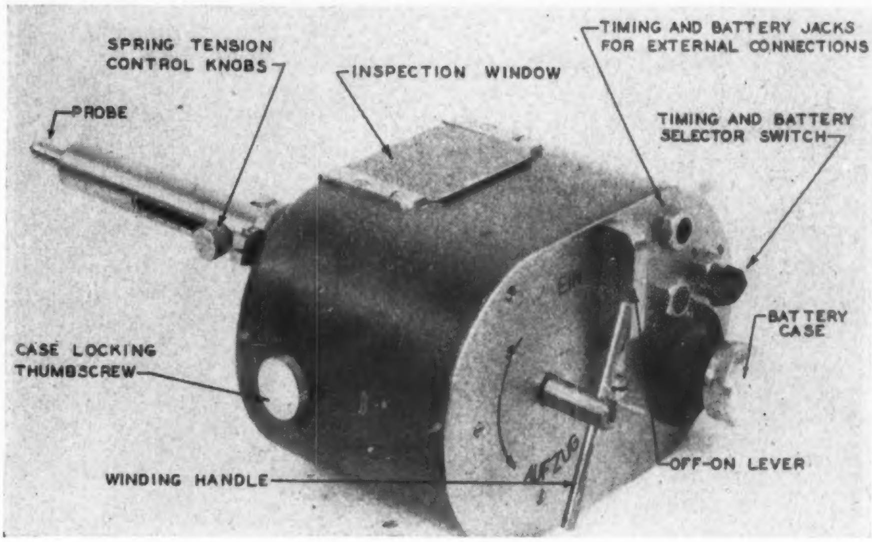


Fig. 1—Right—Simple in design, this mechanical vibration recorder handles frequencies from 5 to 150 cps



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Mechanical Vibration Recorder

. . . achieves practical results through simplicity of design

CAPABLE of handling frequencies from 5 to 150 cycles per second, the mechanical vibration recorder shown in *Fig.* 1 records directly on a moving waxed-paper strip and permits excursions up to one inch.

As the schematic of *Fig.* 2 shows, in operation, motion of a probe is mechanically amplified by a lever which in turn scribes on the moving strip of waxed paper. The paper is pulled from a roll by a spring-wound, governor-controlled motor and stored in a compartment until removed. To provide a frequency reference, timing marks at one-second intervals are made on the waxed paper by the armature of a magnet which is energized by a flashlight cell and controlled by cam contacts in the governor mechanism. In addition, a second timing marker is incorporated in the instrument and is actuated by either an external battery or contacts. A selector switch provides for four combinations of internal and external batteries, and internal and external timing contacts. On the top of the instrument, a Lucite inspection window, *Fig.* 3, is covered by a piece of polished chrome-plated metal, hinged at one end so that it can be used as a mirror in difficult positions. Another Lucite dust cover, located at the bottom of the instrument, can be pulled out for removal of the reel of

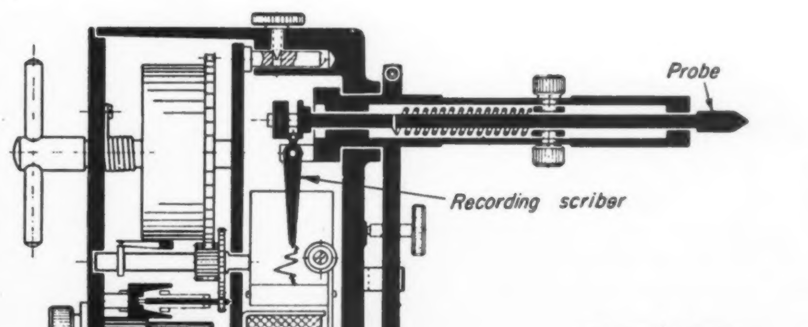


Fig. 2—Right—Schematic shows recorder design and function

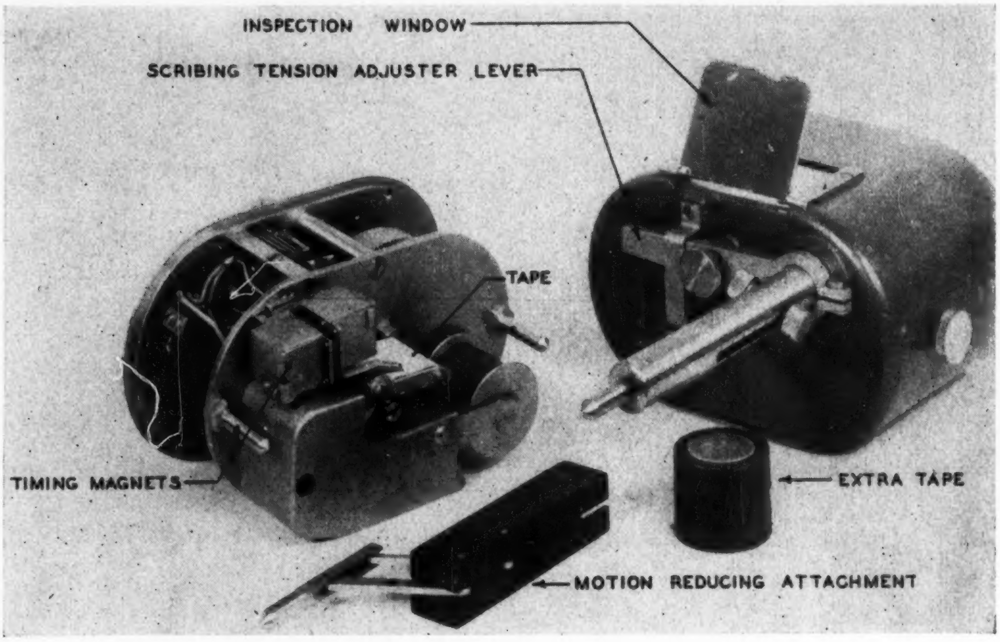


Fig. 3—Recorder partially disassembled, showing key components and construction

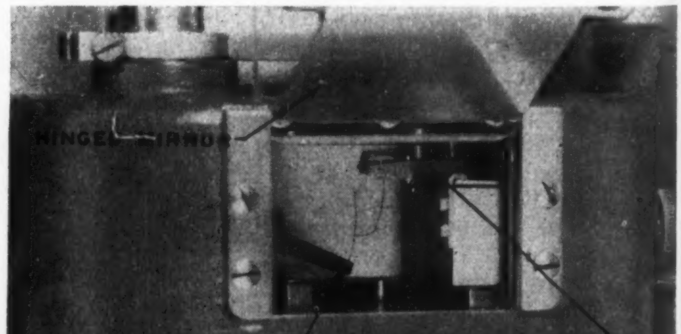
wax paper which constitutes the record.

When hand held, the instrument is essentially seismic for all frequencies above five cycles per second. The upper frequency limit is determined by the force exerted by the compression spring, and maximum spring force will permit frequencies of 150 cps at 20*g* acceleration to be measured. The waxed paper, supplied in rolls nine meters long, travels in the instrument at approximately forty millimeters per second and its driving motor will run for one full minute without rewinding. Mechanical magnification of the vibration amplitude is six while maximum excursion that can be recorded is four millimeters or approximately 0.15-inch. Excursions as low as 0.002-inch can be recorded. With the mechanical reducer attached, a total mechanical magnification up to two can be produced, depending upon the adjustment of the reducer.

It will be apparent that the record is immediately visible and can be analyzed without the use of a microscope or special projection equipment. This is of great value during field measurements where it is desirable to study data on the spot. The mechanical reducing attachment is, of course, used where large vibration amplitudes are encountered. The reducer is slipped over the end of the probe arm and the record obtained immediately. Because of the nature of its design, the scriber arm describes an arc instead of a straight line on the record for a linear movement of the probe, *Fig.* 4, so that detailed analysis of the vibrational wave form is not practical. However, for quick, on-the-spot field measurement of vibrational amplitudes and frequencies, the instrument is highly satisfactory.

It was at the close of the war with Germany that several of these mechanical vibration recorders, made by the Askania Werke, A.G., in Berlin, were taken from a surrendered submarine. They have since been used at the Naval Research Laboratory in connection with many vibration problems. The American model of the instrument will differ in only a few respects. The waxed paper width will be one inch instead of 25 millimeters, and the timing and battery jacks and cables for external connections are to be eliminated. To provide timing, the internal timing mechanism actuated by the governor and enclosed flashlight cell will be retained. Weight of the vibrograph itself is 3.75 pounds. The complete instrument, mounted in its carrying case and with all attachments, weighs 6.5 pounds.

Fig. 4—Below—View through top window reveals that scriber describes arcs instead of straight lines on waxed-paper record



Highlights from ASME Machine Design Division Meeting

KEEN interest in the activities of the new Machine Design division of the American Society of Mechanical Engineers was evidenced by the large attendance at the sessions sponsored by the division during the recent annual meeting of the society in Atlantic City. In addition to a panel session on "Design for Production" under the joint auspices of the Machine Design and Production Engineering divisions, two other sessions were held at which five technical papers were presented. Three of these are condensed on the following pages, and the other two will be abstracted in subsequent issues of MACHINE DESIGN.

Dynamic Forces in Cam Mechanisms

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IN MANY cases the type of motion used to produce the desired amplitude of motion of a cam follower is unimportant in terms of the desired output displacement. In almost all cases, however, the time function employed plays a tremendously important role in the resulting forces produced and therefore greatly influences the life, smoothness of operation, and the permissible operating speed of the machine.

In order to develop the analytical results with which the behavior of the cam-follower mechanism may be predicted, it is necessary to develop the geometric and kinematic properties of the system for a desired follower motion. In this article the following assumptions will be made:

1. The follower is positively driven by the cam
2. No backlash exists in the system
3. The driven output mass is large compared to the other moving components of the cam-driven system
4. The cam rotates at constant angular speed
5. The cam contour is not changed by the forces which act upon it
6. Coulomb friction may be combined with viscous friction and the combined effects presented by an equivalent viscous damping.

Motion of the cam follower is used to impart motion to an output of finite mass. The follower may drive the output member directly or, more commonly, through a system of gears and levers. The follower is coupled to the output member by a relatively low-mass, low-inertia and, in general, relatively stiff system. Generally the mass of the output member is large relative to the follower and the components of the coupling system. Thus no appreciable error results by assuming the mass of all other elements

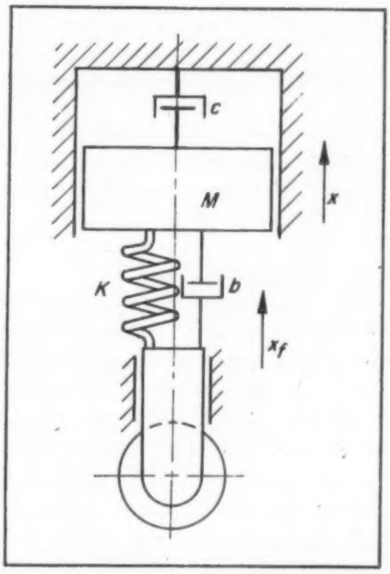
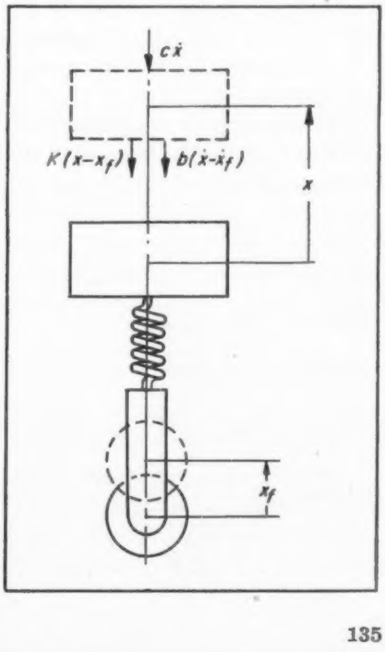


Fig. 1 — Above — Simplified system which is dynamically equivalent to an actual system provided proper values are assigned to the constants

Fig. 2—Below—Follower and output mass at time t=0 (solid lines) and at some time t=t during the motion



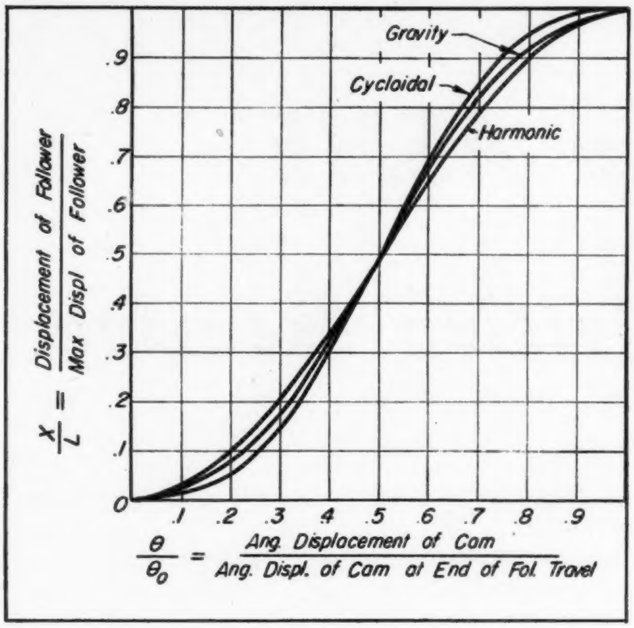


Fig. 3—Displacement characteristics of cam followers with three different types of motion

of the system to be zero.

The equivalent system stiffness, K, is defined as the force required to deflect the follower one inch if the output member is rigidly held, the force being applied at the center of the roller and along the line of motion of the follower. M is equivalent mass of the driven member referred to the displacement x. Its value in any given system is readily found by tracing the inertia force or torque back through the coupling system to the reciprocating motion x. The system shown in *Fig.* 1 is dynamically equivalent to an actual system providing M and K bear the values indicated in the foregoing.

Force transmitted to the follower is the sum of the force developed in the equivalent spring K, and the dashpot force. To determine the maximum cam load it is therefore necessary to find this force as a function of the time for a given follower displacement $x_f = \phi(t)$. The solid lines in *Fig.* 2 show the equivalent mass M and the follower at rest at the time $t = 0$. At some time t the positions are as indicated by the dotted outlines.

A summation of the forces acting on mass M, plus a number of mathematical steps which are given in the original paper, yields the following result:

$$\frac{F}{M} = \omega_n \left[\frac{2\zeta D^2 + (4\zeta_c\zeta + 1)D + 2\zeta_c}{D^2 + 2(\zeta + \zeta_c)D + 1} \right] \frac{dx_f}{dt} \quad (1)$$

where F is the component of force action on the cam follower in the x direction; $\omega_n = \sqrt{K/M}$ = undamped natural frequency of output mass; $\zeta = b/(2\sqrt{KM})$ = damping ratio (follower); $\zeta_c = c/(2\sqrt{KM})$ = damping ratio (frame); b and c are the equivalent viscous damping factors, assuming static and viscous friction may be lumped together with sufficient accuracy and expressed in terms of viscous damping; $D = (1/\omega_n)\ d/dt$.

Choice of a suitable follower acceleration-time characteristic is of great importance. A number of different functions are used, some of the more widely used of which follow:

1. *Constant Acceleration or Gravity Cam.* A cam which produces a follower motion

$$\frac{x_f}{L} = 2\left(\frac{\theta}{\theta_o}\right)^2 \quad (2)$$

2. *Harmonic Displacement Cam.* A cam which produces a follower motion

$$\frac{x_f}{L} = \frac{1}{2}\left[1 - \cos\frac{\pi\theta}{\theta_o}\right] \quad (3)$$

3. *Cycloidal Cam.* A cam which produces a follower motion

$$\frac{x_f}{L} = \frac{1}{\pi}\left[\frac{\pi\theta}{\theta_o} - \frac{1}{2}\sin\frac{2\pi\theta}{\theta_o}\right] \quad (4)$$

The velocity and acceleration characteristics of the follower for the foregoing displacement functions are shown in Table I. Displacement and acceleration curves of the follower are plotted in *Figs.* 3 and 4.

Follower of the constant acceleration cam has the lowest peak acceleration but starts off with a finite acceleration which remains constant until half-lift position is reached, where it suddenly changes to the same negative value. A sudden change back to zero acceleration occurs at full-lift position.

Acceleration of the follower of the harmonic displacement cam starts off with its peak value and changes continuously, passing through zero at half lift. At full lift, the acceleration undergoes a sudden change from its peak negative value to zero.

Acceleration of the cycloidal cam changes continually from zero at the start of lift to zero at full lift, exhibiting no sudden changes. Its peak value is higher than with the previously discussed cam contours.

Table I—Follower Characteristics

Type of Cam	Displacement	Velocity	Acceleration
Constant acceleration....	$\frac{x_f}{L} = 2\left(\frac{\theta}{\theta_0}\right)^2$	$\frac{v_f}{L\omega_f} = \frac{4\theta}{\theta_0^2}$	$\frac{a_f}{L(\omega_f)^2} = \left(\frac{2}{\theta_0}\right)$
Harmonic displacement..	$\frac{x_f}{L} = \frac{1}{2}\left[1 - \cos\frac{\pi\theta}{\theta_0}\right]$	$\frac{v_f}{L\omega_f} = \frac{\pi}{2\theta_0}\sin\frac{\pi\theta}{\theta_0}$	$\frac{a_f}{L(\omega_f)^2} = \frac{1}{2}\left(\frac{\pi}{\theta_0}\right)^2 \cos\frac{\pi\theta}{\theta_0}$
Cycloidal	$\frac{x_f}{L} = \frac{1}{\pi}\left[\frac{\pi\theta}{\theta_0} - \frac{1}{2}\sin\frac{2\pi\theta}{\theta_0}\right]$	$\frac{v_f}{L\omega_f} = \frac{1}{\theta_0}\left[1 - \cos\frac{2\pi\theta}{\theta_0}\right]$	$\frac{a_f}{L(\omega_f)^2} = -\frac{2\pi}{\theta_0^2}\sin\frac{2\pi\theta}{\theta_0}$
		$\left(v_f = \frac{dx_f}{dt}\right)$	$\left(a_f = \frac{d^2x_f}{dt^2}\right)$

If the foregoing follower motions are inserted in Equation 1, the solution of the resulting differential equations may readily be obtained. The results follow:

For constant acceleration cam

$$\frac{dx_f}{dt}=a_gt; \quad a_g=4L\left(\frac{\omega_f}{\theta_o}\right)^2$$

Substituting in Equation 1 gives the following result:

$$\frac{F}{Ma_g}=\left[\frac{2\zeta D^2+(4\zeta\zeta_c+1)D+2\zeta_c}{D^2+2(\zeta+\zeta_c)D+1}\right]\omega_n t \quad \ldots\ldots\ldots\ldots (5)$$

For harmonic displacement cam

$$\frac{ax_f}{dt}=\frac{a_h\theta_o}{\pi\omega_f}; \quad a_h=\frac{L}{2}\left[\frac{\pi\omega_f}{\theta_o}\right]^2$$

By substituting in Equation 1

$$\frac{F}{Ma_h}=\frac{\theta_o}{\pi\beta}\left[\frac{2\zeta D^2+(4\zeta\zeta_c+1)D+2\zeta_c}{D^2+2(\zeta+\zeta_c)D+1}\right]\times$$

$$sin\left(\frac{\pi\beta}{\theta_o}\right)\omega_n t \quad \ldots\ldots\ldots\ldots (6)$$

where $\beta=\omega_f/\omega_n$.

For cycloidal cam

$$\frac{dx_f}{dt}=\frac{\theta_o a_c}{2\pi\omega_f}\left[1-\cos\frac{2\pi\theta}{\theta_o}\right]$$

where

$$a_c=2\pi L\left(\frac{\omega_f}{\theta_o}\right)^2$$

By substituting in Equation 1

$$\frac{F}{Ma_c}=\frac{\theta_o}{2\pi\beta}\left[\frac{2\zeta D^2+(4\zeta\zeta_c+1)D+2\zeta_c}{D^2+2(\zeta+\zeta_c)D+1}\right]\times$$

$$\left[1-\cos\frac{2\pi\beta}{\theta_o}\omega_n t\right] \quad \ldots\ldots\ldots\ldots (7)$$

Solutions to Equations 5, 6, and 7 have been carried out on the Differential Analyzer at M.I.T. for selected values of the system constants. The results are plotted in *Fig.* 5.

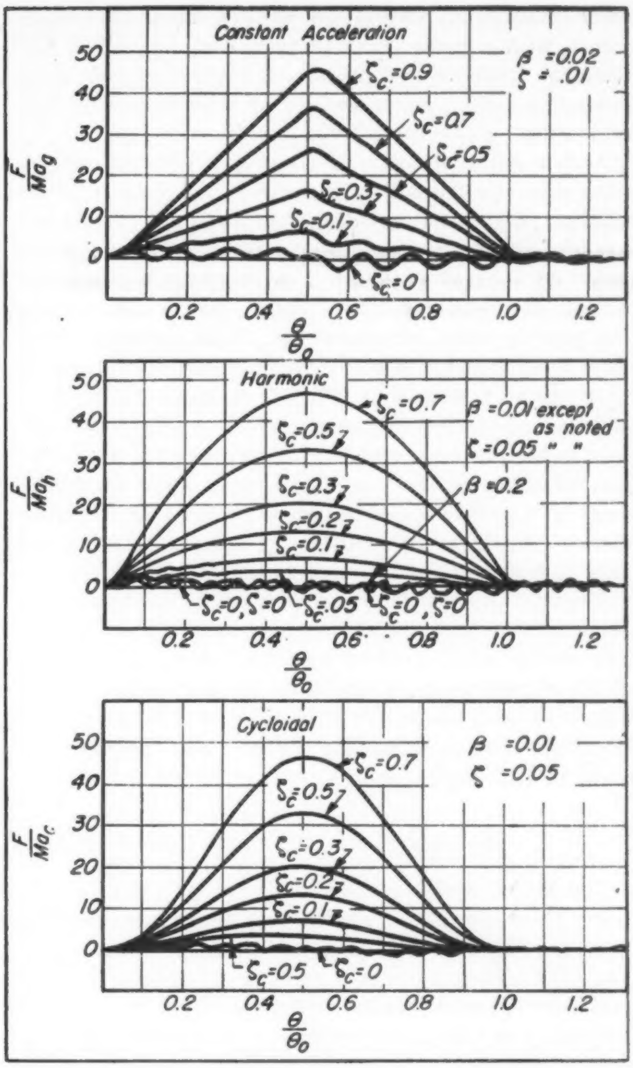
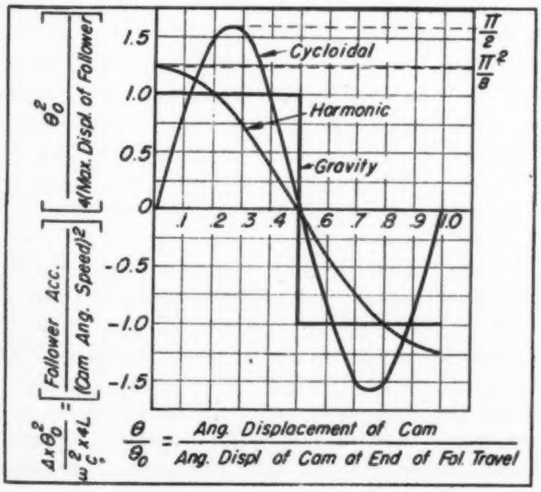


Fig. 5—Ratios of axial cam force to product of driven mass and maximum follower acceleration, plotted against ratio of cam displacement required to produce full lift

Ratio of the force on the cam along the center line of the follower to the product of the equivalent output mass and the maximum acceleration of the follower is plotted against the ratio of cam displacement to the cam displacement required to produce full lift. This product, Ma, is the maximum value F would reach if the motion of the equivalent mass were the same as the cam follower. This ratio is therefore a measure of the cam force amplification resulting from the spring-dashpot coupling between the cam follower and the output. In each case the motion plotted occurs between two dwell periods.

With no damping present, the force F oscillates at the natural frequency (ω_n). In the case of the gravity cam the peak value of the ratio F/Ma_g during the first half of the follower motion is equal to two. Immediately after midposition where the follower acceleration suddenly changes from a positive value to a negative one of equal magnitude a peak force ratio of three is reached, *Fig. 5a*. With a harmonic fol-

Fig. 4—Acceleration characteristics compared for the three types of motion shown in Fig. 3



lower motion an oscillation of the same frequency occurs with a peak value of two, *Fig. 5b*. With the cycloidal follower motion, an oscillation of the same frequency but of much smaller amplitude occurs. The peak value of the force ratio is 1.06, *Fig. 5c*.

As the damping ratio ζ_c (damping between output mass and the frame) is increased, the transient oscillation decreases rapidly but the peak force increases rapidly with damping ratio. For damping ratios (ζ_c) greater than 0.1 with cycloidal follower motion no force reversal occurs. For the harmonic and gravity cams a very slight force reversal takes place near the end of the travel for values of ζ_c equal to or greater than 0.1. This is because of the fact that the damping forces are now large relative to the inertia forces. As the damping ratio increases, the general shape of the force-time curve approaches the shape of the velocity-time curves of the cam follower. Values of ω/ω_n were arbitrarily chosen and the results are shown for two values of β. Normally the natural frequency of the driven system is high relative to the frequency of rotation of the cam. Therefore ratios of $\beta = 0.01$ and 0.02 were used. The effect of changing values of β in the range of low values of β is shown in *Fig. 5b*. In comparing the plotted results it should be remembered that a_c/a_g is equal to 1.57 and a_h/a_g is equal to 1.24.

Advantages of Cycloidal Cam

The use of a cam contour of the cycloidal type produces the following desirable results at low damping ratios: (1) A lower peak force along the axis of the cam follower; and (2) The amplitude of the transient force variation taking place at the natural frequency of the driven system is much smaller than that produced by the other two cam contours. This should result in longer life of the cam surface and other elements of the machine. It should yield quieter as well as smoother machine performance. Operation at increased speeds should be possible.

In order to produce a cycloidal follower motion the accuracy to which the cam contour must be held is very high. An error of a few ten-thousandths of an inch in the initial or final stages of the rise will seriously affect the system performance. This is also true in the case of gear action. In recent years a tremendous improvement in gear performance has resulted from the recognition of need for holding tooth contours within very close limits. A great improvement in cam operation will result when the importance of close tolerances is fully recognized.

It will well to point out that the cycloidal follower displacement does not necessarily represent an optimum solution to the cam problem. However, it does offer an opportunity for tremendous improvement in dynamic performance over that obtained by either the gravity or harmonic displacement type cams. A dynamic analysis of the results produced by any other cam contour can be carried out by the methods outlined in this article. The idealized treatment used in the foregoing has omitted the influence of a number of important factors (see list of assumptions) which may be introduced in future work.

Ball Bearing Slides

By Conrad Jobst

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FOR straight-line movements, friction slides are still common and in a few cases V-type roller bearings are applied to machines such as thread grinders, etc. However, on larger machine tools friction slides are universally used. It should be of interest to relate the author's experiences with a preloaded ball bearing slide which has been shown in other patents but has itself never been patented.

As a background for comparison, *Fig. 6* shows conventional slides which must be provided with adjustments for wear and oil clearance. All loads must be arranged near to the center of the slide if cocking and binding are to be prevented. The slides are necessarily loose in the conventional sense in order to permit free motion, and if any additional load is applied, a certain amount of instability must exhibit itself, that is, instability in the realm of extremely accurate

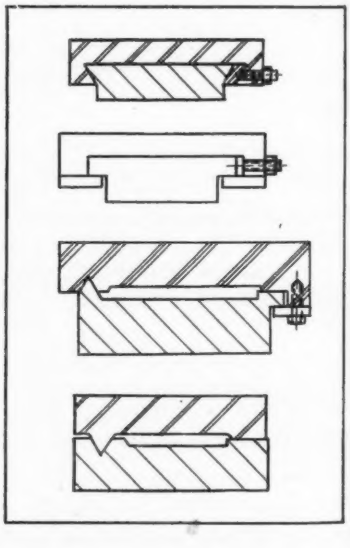

Fig. 6—Typical conventional slides showing provisions for adjustment to compensate for wear and oil clearance

motions. For those not familiar with such accuracy, an example would be the change in location of a cutting tool due to change in thickness of the oil film in the slide. The simplest procedure to get stability into a mechanism is to preload the entire assembly, and this is accomplished by designing a rolling contact between two slide members, with the initial preload so large that subsequent loads are small by comparison.

To overcome these serious objections of instability in motions supposedly accurately controlled, the author replaced the sliding surfaces with preloaded ball bearing tracks as shown in the two diagrams of *Fig. 7*. Preloading is produced in the desired amount by control of the tolerances of the slide members prior to assembly. Manual adjustment subsequent to assembly is omitted purposely. This particular mechanism

is the cross slide on a machine in which the alignment between two-positioned operations must be kept perfect in order to make a high-grade product. It was found that this accurate alignment due to the preload principle has actually been maintained for over 20 years of operation, and the original machine is still in operation.

In *Fig.* 8 is shown the complete unit which is composed of two tracks and spacer with steel balls. Dimensions *A* and *B* may be kept standard for interchangeability. There are many methods of dirt protection for such slides, which depend on the location of the slide. The position of such a slide in a design may be chosen for convenience, because any eccentric or off center load does not impair its precise movement, and cocking of the ball bearing slide in the sense of cocking of a conventional slide is impossible.

In *Fig.* 9 is shown the assembled slide unit; *B* is the male and *A* is the female slide running on track units. Dimension *D* is made oversize, and therefore establishes the preload, or prestress, desired. From

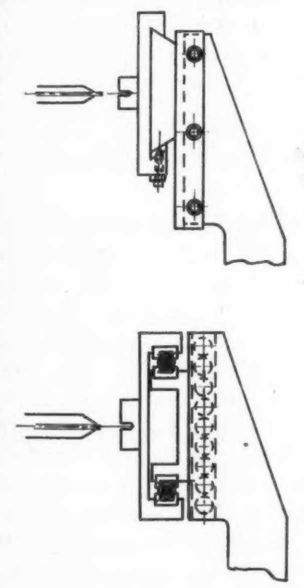

Fig. 7 — Left—Application of preloaded ball bearing tracks to two slides on a precision machine

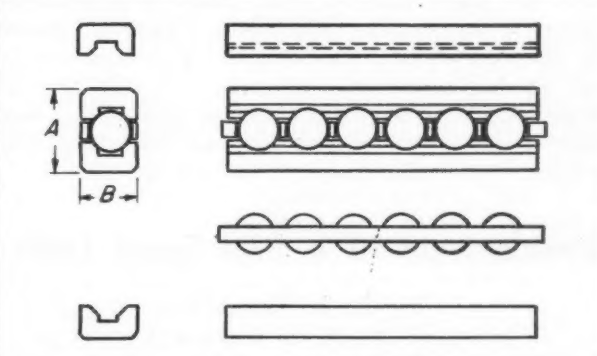

Fig. 8 — Right—Component parts of ball bearing slide, consisting of two tracks, spacer and steel balls

a practical standpoint, dimension B_1 for both slides should permit a press fit for the tracks.

An example of heavy machine design is a mold-clamping device with 1,000,000 lb clamping pressure, but having a sensitivity that makes complete mold protection possible. It takes only 150 lb spring pressure to move the carriage which weighs 5,000 lb, including the die. The preload on all the balls is approximately 50 tons. The meeting of the die or mold faces is extremely accurate and does not depend upon pin and bushing, in fact, no pins are used at all. Because of the absolutely parallel meeting of the die faces, and because of the free movement in the slide, mold protection possibilities suggested themselves.

If foreign matter thicker than 0.001-in. lodges between the die faces, particularly parts left over by flashes of the plastic material, or if the whole molded piece is left in the mold, the carriage cannot complete the entire stroke, so that the micro safety switch is not opened, and the springs compress due to the forward motion and actuate a switch to reverse the motor and prevent pressure being applied to the die faces. It prevents closing the switch which controls the torque unit gear motor. This torque gear mechanism applies 500 tons pressure to the mold faces, and that pressure would never be created if the mold faces are separated more than 0.001-in. Such prcision is not possible with conventional slides. It has been found that a cigarette placed between the mold faces has sufficient compressive strength to actuate the spring without breaking the paper. The cigarette did not resist the whole 150 lb required to move the carriage, but it did build up sufficient resistance to produce thrust against the forward movement which, being free, would yield and actuate a micro switch. The placing of a 0.001-in. shim or tissue paper at any part of the mold face over a 20 sq in. area would actuate the micro switch, proving that the lateral movement of the carriage is negligible.

This particular molding machine has been in operation for 3½ years. A test revealed that it took the same twisting force against the platten to show deflection reading after 3½ years as it did after it was three months old. This comparison was made after the initial three months, rather than when new, to permit plastic flow, or "cold flow," to become negligible.

The principles of the straight-line preloaded ball bearing can be applied to the helix, and thereby obtain similar results in rotary motions. To provide power for the closing of a mold and injection of the plastic material into the mold, a helical ball bearing was used which creates pressure by rotation (a screw). In operating these two units (the screw and the slide) for molding thermoplastic resins, it was found that the cost of electric current for the 10-hp motor to operate them was less than the cost of cooling the oil for the hydraulic machine of the same capacity, which required a 35-hp motor.

Motors of surprisingly small horsepowers are fixed to either or both rotating members which, with proper

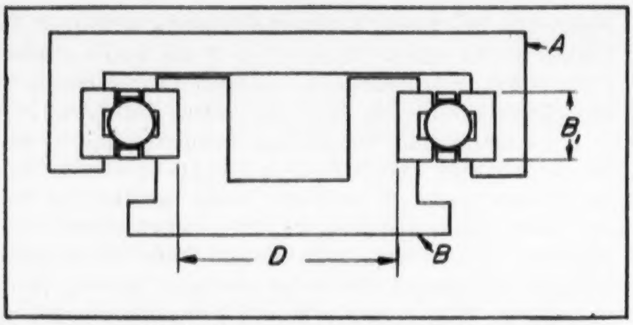

Fig. 9—Assembled slide unit. Dimension "D" is made oversize in order to provide the desired preloading

electrical controllers, can be cycled automatically, or by manual control, for continuous production. Excellent sensitivity is obtained. The pressure output is many times greater than for a hydraulic system, and this higher pressure is obtained by the short-time overload which the motors can withstand and will provide. In any hydraulic system, oil pressure can be no greater than that for which the pumps were designed, not even for short intervals.

The work of final filling of the cavity is done by the inertia of the element. This accounts for the small use of power. The element remains stationary, much the same as a nut once tightened requires no pressure to hold it locked. The functions of these parts are quite different from those of a hydraulic system which require continuous pumping against the pressure induced in molding. This pressure by ball bearing screw can be applied to any type of press. The general overall dimensions of such a machine would be small in comparison to a hydraulic machine of the same power.

Development of a High-Speed Lathe

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PRACTICAL limitations on the maximum machining speeds for aluminum so far have been imposed by the limitations of the machine tools rather than by the machining properties of aluminum.

Milling machine spindles have been operated experimentally at speeds ranging from 6000 to 9000 rpm, corresponding to cutting speeds of 5000 to about 20,000 ft per min, using the high-cycle type of motor drive. Under such conditions of operation the spindle bearings failed rather quickly because they were not designed for these conditions. As would be expected it was found that higher spindle speeds required appreciably more horsepower than normally provided for the machine tools. It seems evident, therefore, that if the advantages of the excellent machining properties of aluminum are to be realized, the maximum speeds of machine tools must be increased considerably. This will require not only more power but also consideration of many other factors that will be affected by the higher speeds.

Because of its interest in extending the use of higher cutting speeds for machine operations on aluminum, the Aluminum Company of America, in 1944, decided to initiate the development of an experimental high-speed lathe capable of doing turning operations at speeds well beyond those currently available in conventional commercial lathes. This development has been pursued with the co-operation and assistance of leading manufacturers of machine tools, bearings, electrical equipment, and cutting tool materials.

In the development of an experimental machine of this type it was necessary first to select the size of the machine to be designed and built. After consideration of the many factors involved, it was decided to develop a turret-type lathe with a spindle having an internal bore of about 2½ inches which would operate at a maximum speed between 7500 and 10,000 rpm. Two serious problems were immediately evident: What kind of bearings and what type of drive should be used for the machine? Consideration was given to ball, roller, and plain bearings before deciding to use the Filmatic segmental type. Although bearings of this type and of the size needed had never before been operated at the top speed specified, the manufacturer thought this type of bearing would function satisfactorily under the conditions indicated, with the result that these bearings were selected for both the radial and thrust bearings of the lathe spindle and the radial bearings of the motor.

Spindle Bearings Hydraulically Adjusted

In the design and making of the spindle and motor bearings, and the spindle and motor housings, Cincinnati Milling and Grinding Machines Inc. co-operated. They also made the motor shaft and assembled the special motor rotor on it. A longitudinal section of the spindle is shown in *Fig.* 10. The motor bearings are lubricated with oil from a central system at 4 psi pressure. The motor bearing shoes are of the fixed type and are arranged with 0.005-inch diametral clearance on the shaft. The bearing shoes for the spindle are adjusted by hydraulic pressure which is regulated by a manually operated control valve. An oil pressure of 18 psi is maintained in the bearing chamber of the spindle housing for lubrication of the shoes.

It was recognized initially that suitable provision would have to be made for controlling the temperature of the lubricating oil used for the spindle and motor bearings. Therefore, a 5-hp hermetically sealed Air-Temp cooling unit was used to control the temperature of both the lubricating oil and the hydraulic oil for operating the feeding mechanism. A centrifugal pump was attached to the oil storage tank suction line to circulate the oil through the Air-Temp unit and return it to the tank.

Various types of drive including steel belt, V-belt, roller chain, gear, and direct connected were considered initially, the last named being the final choice. Selection of this type of drive increased the complexity of the electrical equipment considerably but did provide a workable solution to the immediate problem presented. The high speed of the spindle with a

direct-connected motor indicated a squirrel-cage type motor, which was adopted. The motor was designed with a hollow shaft having an internal bore corresponding to that of the lathe spindle.

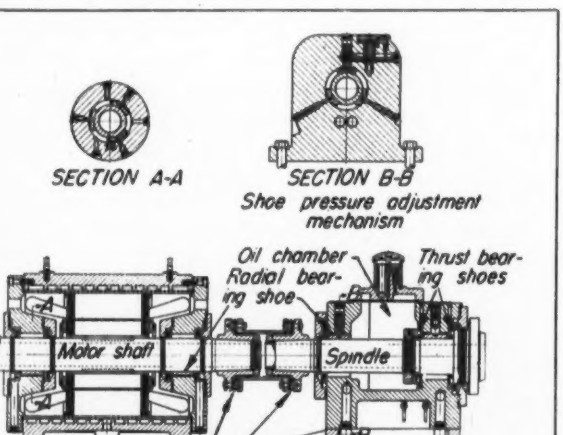
The spindle motor, which was furnished by the Reliance Electric and Engineering Co., has a rating of 80 hp at 220 volts, 3 phase, 60 cycle, 3450 rpm. This same horsepower rating is maintained as the frequency is increased to 150 cycles, which corresponds to a top speed of about 8800 rpm, and during this frequency and speed increase the voltage is raised to a maximum value of 350 volts. In order to handle the additional high losses resulting from starting and stopping, the stator is provided with water cooling.

Since the project was entirely experimental, a power supply of one or more fixed frequencies was not considered suitable, as large gaps in the test data would result. Because of this, a rather elaborate electrical set-up was provided, *Fig.* 11, involving, in addition to the spindle motor, two motor-generator sets. One of these sets is a conventional, constant-speed, ac-dc set which provides a variable d-c voltage from zero to 250 volts. The variable voltage supplies the motor of a second set comprising a d-c motor and a-c alternator which can be operated at any speed between zero and 1800 rpm, so that the lathe spindle can be operated at any speed desired between zero and 8800 rpm.

One of the main problems was to provide a voltage input to the spindle motor proportional to speed from zero to 3450 rpm and then vary the voltage on a straight-line basis from 220 volts at 3450 rpm to 350 volts at top speed. Amplidyne control provided this voltage control and incorporated voltage forcing during periods of heavy demand on the alternator, such as exists during the accelerating period of the spindle motor.

The other control problems involved the provision of relatively quick starting and stopping of the spindle motor. During such starting and stopping the speed of the alternator motor-generator set is not varied,

Fig. 10—Longitudinal section through spindle and motor, showing segmental-type journal and thrust bearings



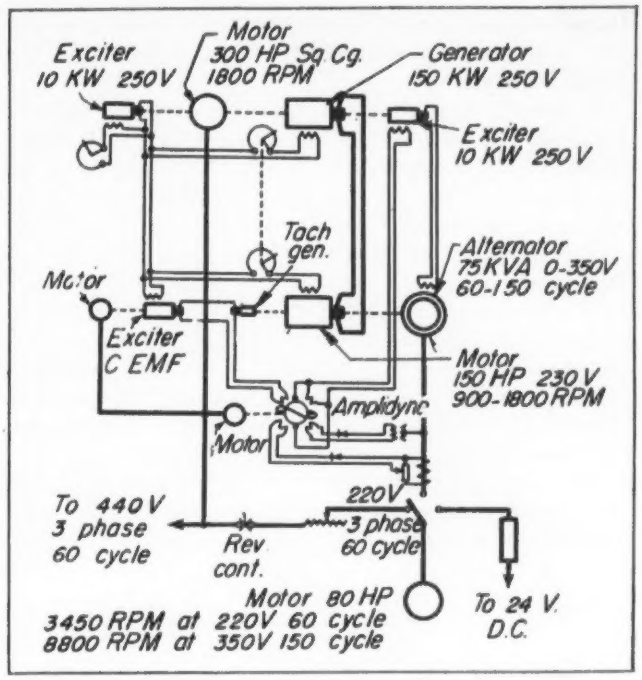


Fig. 11—Electrical circuit for providing infinitely variable speed changes up to 8800 rpm

and when the motor is started it will quickly come up to whatever speed is represented by the preset speed of the alternator set.

In starting, for ultimate speeds above 3450 rpm the spindle motor is thrown across the 220-volt, 3-phase, 60-cycle line in the conventional manner, and left there until it attains a speed of slightly below 3450 rpm. At this point the motor is transferred to the alternator and simultaneously the field of the alternator is forced during the remainder of the acceleration period. The motor then continues to accelerate to whatever speed corresponds to the preset frequency of the alternator set.

Assuming that the motor is operating at some speed above 3450 rpm, the stopping sequence consists in first disconnecting the motor from the alternator and connecting it to the 220-volt, 3-phase, 60-cycle line, the rotation being such that regeneration (not plugging) occurs. This quickly slows down the motor, and when a speed slightly above 3750 rpm is reached, the motor is disconnected from the 60-cycle line, and direct current of approximately full-load value is sent through two legs of the stator winding. When the motor comes to a stop, this direct current is disconnected.

An alternative method of stopping employs plugging from the 60-cycle line for the deceleration below 3600 rpm. This has been tested with excellent results, but required the additional complication of a zero speed switch, therefore most of the operation has been done with the direct-current method of deceleration and stopping. A stop from full speed requires between four and five seconds, and a similar time is required for acceleration.

The operator's control consists basically of a start pushbutton, a stop pushbutton, and a rheostat which may be preset for any operating speed; of course,

this speed may be changed while a cut is being made. The controls were furnished by the Electrical Controller and Mfg. Co.

It should be pointed out that most commercial applications of such a high-speed machine tool would require only a comparatively simple electrical set-up providing for one or two fixed frequencies. Such a scheme would be quite economical and might provide, for instance, for two operating speeds, say 3600 rpm and 7200 rpm. Quite confidently it may be expected that where a complete range of variable speeds is required, the electrical industry will within the next few years be able to provide an electronic frequency changer as a power source, which will eliminate a great deal of the expensive and bulky equipment used for this experimental installation.

The motor was connected to the back end of the spindle by means of a dynamically balanced, forged aluminum alloy Thomas flexible coupling. The ends

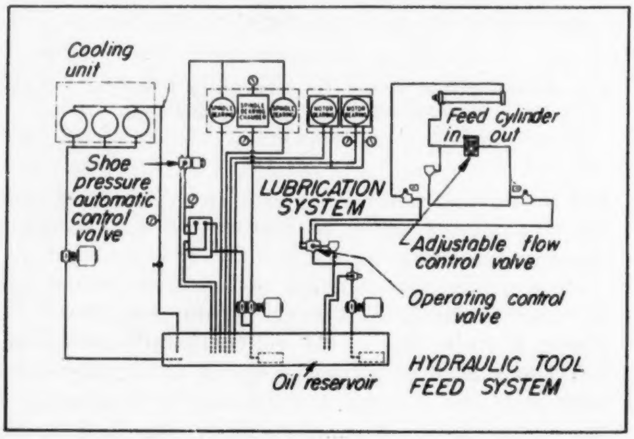


Fig. 12—Diagram of hydraulic circuit for feed control

of the coupling were pushed on the tapered ($\frac{1}{2}$-inch per ft on diameter) ends of the spindle and motor shafts and retained by spanner nuts. No keys were provided and the friction of the tapered joints has proved quite satisfactory under the maximum torque loads used.

A hydraulically operated tool carriage with a separate hydraulic circuit, *Fig.* 12, and Vickers pump for operating the carriage longitudinally was provided. A manually controlled operating valve mounted on the side of the spindle housing controls the carriage movement through a suitable hydraulic cylinder. Cam-operated shut-off Vickers valves were provided in the same circuit for stopping the carriage at any desired point in either direction, within its maximum range. A manually operated cross-slide and a turret type tool holder were mounted on the tool carriage.

Advance feed speed of the tool carriage is controlled by a Vickers flow-control valve mounted on the front of the machine bed and a rapid return speed is obtained by allowing full-line oil volume to pass to the cylinder when the operating valve is set for return. A maximum feed of 120 in. per min was provided. Subsequent to the first tests of the lathe, changes were made in the tool carriage and lathe bed so that the carriage could be fed transversely by the hydraulic mechanism. This arrangement was effected for the purpose of carrying out high-speed milling tests in which the cutter was mounted on the spindle nose and the work clamped on the tool carriage.

A heavy welded steel bed was provided for the machine by Alcoa, on which the motor, spindle housing, and tool carriage assembly were mounted. The interior of the machine bed is used for an oil storage tank which is provided with suitable baffles and filters. The hydraulic pumps and pressure control valves are mounted on the rear plate of the bed.

A collet chuck of $2\frac{1}{4}$-in. maximum capacity was supplied by the Warner and Swasey Co. This chuck is operated manually by a lever mounted on top of the spindle housing. In addition, a special 10-in. three-jaw scroll type chuck was furnished by the same firm. The body of this chuck is made from an aluminum alloy 14S-T forging. Provision is made for bolting this chuck to the spindle nose of the lathe and the chuck jaws are adjusted by the usual type of wrench.

What the Tests Showed

CONCLUSIONS: From the results of tests conducted on the experimental high-speed lathe, in which as-rolled 14S and 14S-T stock and 24S-T plate were machined dry at surface cutting speeds up to 20,000 fpm, a number of conclusions appear to be warranted of which the following are typical:

1. Aluminum alloys can be satisfactorily machined by turning at cutting speeds up to 20,000 fpm. No indication of an upper speed limit was observed

2. The number of cubic inches of metal which can be removed per minute for each horsepower going into the cut is greater for as-rolled 14S than for 14S-T or 24S-T and increases with increasing rate of tool feed. In these tests as much as 7.5 cubic inches of metal were removed for each horsepower developed at the cut

3. The net horsepower required to remove a given volume of metal is independent of cutting speed and depth of cut

4. As much as 470 cubic inches of metal per minute (approximately 47 lb) were removed without overtaxing the equipment. There was no indication that greater amounts could not be removed

5. The amount of power required to drive the spindle when a cut was not being made was greater than had been anticipated. Extrapolation of the data indicate that the 80-horsepower motor might be operating at as much as 70 per cent of capacity when driving the idle spindle at 9000 rpm

6. Increasing the clearance on thrust and radial bearings had little effect on the idle horsepower but varying the hydraulic pressure to the spindle bearing shoe cylinders did cause some variation in this idle power

7. Changing the oil in the lubricating and hydraulic system to a lighter oil resulted in an average reduction in idle horsepower of approximately 17 per cent.

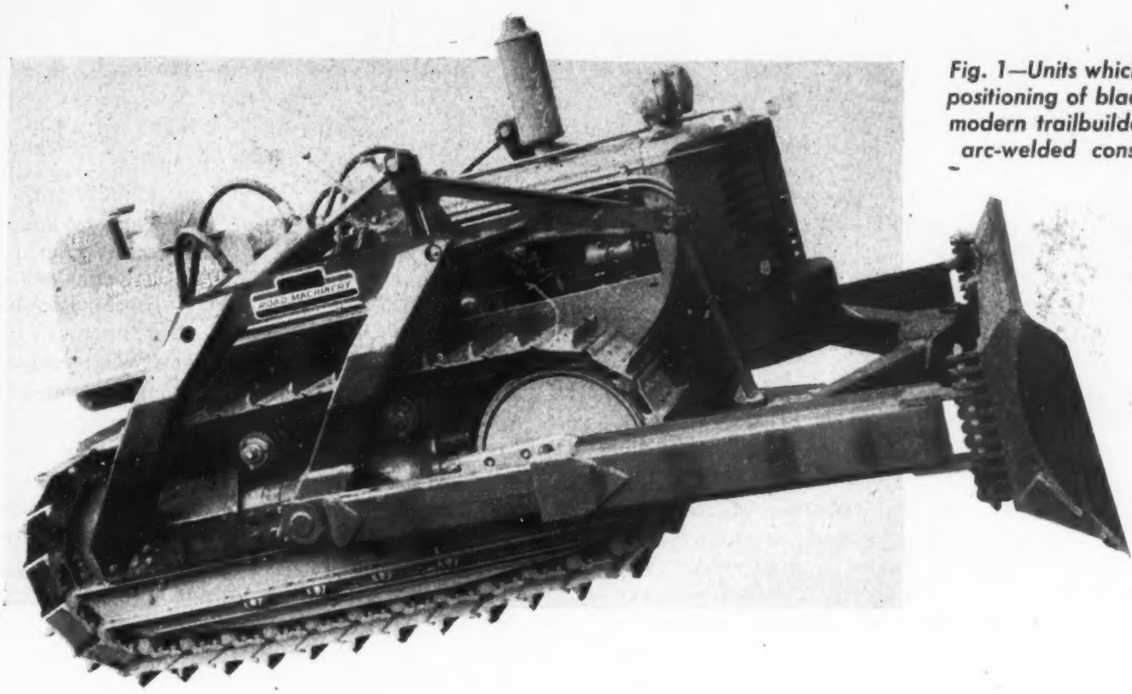


Fig. 1—Units which actuate positioning of blade of this modern trailbuilder are of arc-welded construction

How Trailbuilder Units Were Designed for Arc Welding

By G. J. Storatz

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DESIGN of dozers and trailbuilders was proceeding in line with the times until the Second World War when engineers were called upon to make radical changes in designs to overcome temporary shortages in materials and manpower. There were times when urgently needed equipment had to be fabricated of steel plate because of the inability to obtain castings. Since it was a case of having something rather than nothing to offer the armed services for the successful prosecution of the war, welded fabricated machines, *Fig.* 1, replaced previous cast designs. It is only natural, therefore, that these experiences should be carried over into the design of the commercial machines of the present day.

CONDENSED from a paper which won a major award in the recent "Design-For-Progress Program" sponsored by The James F. Lincoln Arc Welding Foundation, this article discusses the design and fabrication of key units of a modern trailbuilder. Data are also presented on cost estimating of arc-welded structures

Result of this wartime expediency was that the engineer found he could develop a fabricated type of machine which in this case, not only cost less to manufacture, but also resulted in a saving of man-hours, material and weight. Since our shop is primarily a plate shop, it was good economical practice for us to fabricate the majority of the parts of welded plate.

However, it should not be assumed that it is possible at all times to replace castings entirely with fabricated construction. A cast member sometimes can be profitably redesigned to make it of cast *and*

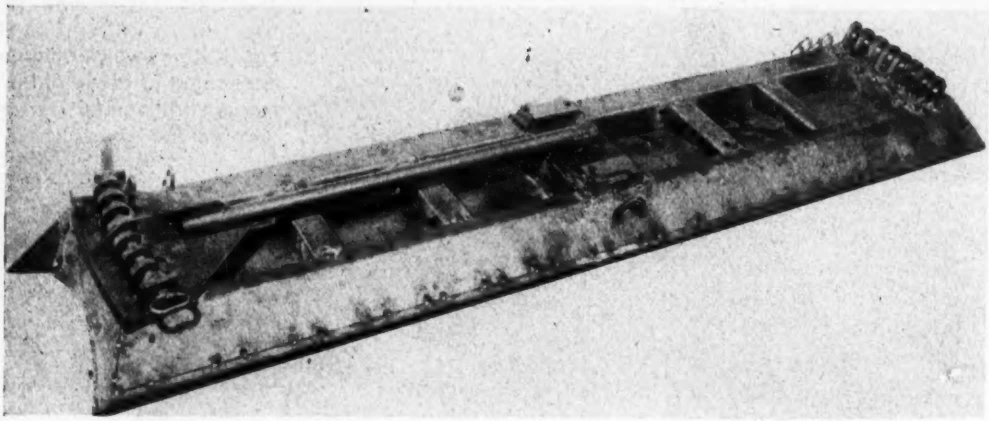


Fig. 2 — Welded construction of moldboard asembly which is subjected to a variety of stresses from simple bending to complex torsion and bending

welded construction. In the redesign of the trailbuilder, a thorough study was made to determine if it was possible to replace major castings with fabricated construction, and it was found that a substantial saving in weight and cost could be realized if the unit was redesigned for welded construction. This saving in weight, when passed on to the contractor, resulted in his being able to move more dirt faster and at a lower cost per yard.

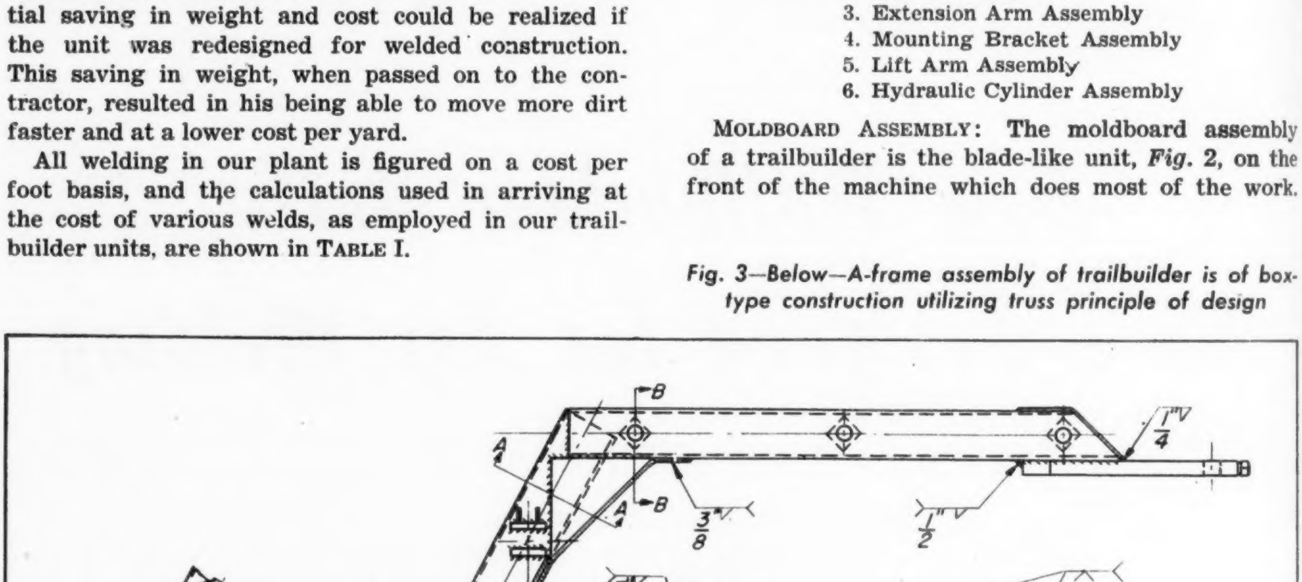
All welding in our plant is figured on a cost per foot basis, and the calculations used in arriving at the cost of various welds, as employed in our trailbuilder units, are shown in TABLE I.

The following is a break-down of the major assemblies of the trailbuilder which were redesigned:

1. Moldboard Assembly
2. A-Frame Assembly
3. Extension Arm Assembly
4. Mounting Bracket Assembly
5. Lift Arm Assembly
6. Hydraulic Cylinder Assembly

MOLDBOARD ASSEMBLY: The moldboard assembly of a trailbuilder is the blade-like unit, *Fig.* 2, on the front of the machine which does most of the work.

Fig. 3—Below—A-frame assembly of trailbuilder is of box-type construction utilizing truss principle of design



It has a curved front face so designed as to impart a rolling action to the dirt. The moldboard can also be tilted so that either end penetrates into the ground for specialized routing work. Across the lower end of the moldboard is a cutting edge which is bolted on, and this is used to obtain penetration and impart a shearing action on the ground. This moldboard is also used for moving rock, routing out stumps, etc.

Since the moldboard bears the brunt of the work, it must be most rigidly constructed. The moldboard front sheet usually is a medium-alloy steel plate both for wear resistance and impact strength. This front sheet is backed up by reinforcing longitudinal and vertical cross members. The moldboard is loaded in such manner that it acts under certain conditions as a beam with a concentrated load when dozing stumps and rocks and, when used for pushing dirt, it acts as a uniformly loaded beam sustaining forces along its entire length. This loading subjects the moldboard to a variation of stressing ranging from simple bending to complex torsion and bending. Various members of the moldboard under operation will sustain a reversal of stresses. Members subject to tension in one cycle of the operation may be subjected to compression during another cycle.

Welded and Cast Designs Similar

In the fabricated design, general appearance of the moldboard was similar to the cast design. The front sheet was also ¼-inch high-tensile plate and the upper and lower tunnel sections were of the same thickness and size as the sections in the cast design. The four vertical stiffeners of the moldboard were also similar in construction and size to the cast design. However, in the fabricated design the center trunnion block was simplified so that a rotary action was accomplished on the back of the blade by tongue and groove construction. Pivotal action was obtained by the insertion of a bearing block between two adjacent ears of the A-frame. This construction resulted in a much simpler manufacturing procedure as well as better balance in the trailbuilder because it was possible to bring the moldboard closer to the tractor.

End adjustment castings were eliminated entirely, and in place of them a box section of ⅜-inch plate was used. On the rear of the box section a series of ears were welded radially in relationship to the center of the blade. The ears of the extension arms meshed with these moldboard ears and were tied to-

TABLE I

Calculation of Welds

Labor	$1.68 per hour
Power	$0.02 per KWH
Electrode	$0.095 per lb
Efficiency of Machine	50 per cent

1/4-INCH BUTT WELD

$$\text{Labor} = \frac{\text{Cost of Labor/hr}}{\text{Welding Speed ft/hr}} = \frac{1.68}{45} = \$0.037$$

$$\text{Power} = \frac{\text{Amp}\times\text{Volts}\times\text{Cost per KWH}}{\text{Efficiency}\times\text{Welding Speed ft/hr}\times 1000}$$

$$= \frac{325\times 33\times 0.02}{58\times 45\times 1000} = 0.008$$

$$\text{Electrode} = (\text{Lb of Electrode/ft})\times(\text{Cost/lb})$$

$$= 0.225\times 0.095 = 0.0214$$

$$\text{Cost of Interruption} = \text{Lb Electrode/ft}\times\text{Labor Cost/lb}$$

$$= 0.225\times 0.0133 = 0.0029$$

$$\text{Overhead} = 200 \text{ per cent Labor Cost} = 0.074$$

$$\text{Total Cost per foot 1/4-inch Butt Weld} = \$0.1433$$

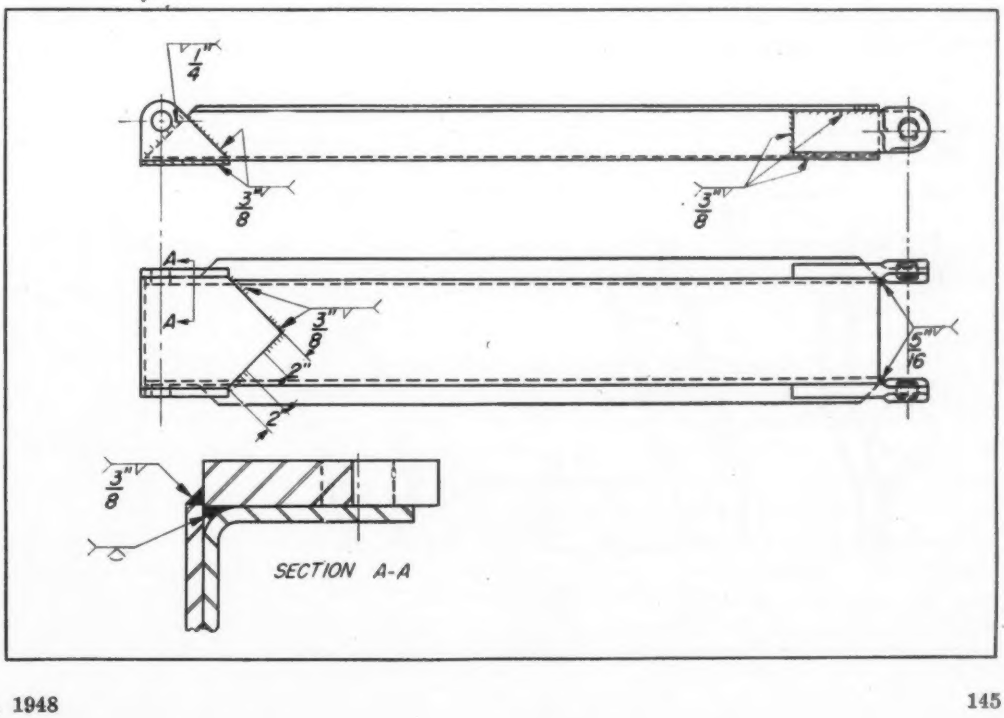


Fig. 4—Extension arm is of flanged channel-shaped plate designed to take loading in compression and tension

gether with a curved T-type of pin. This construction was relatively simple and allowed an appreciable saving in cost and weight.

Fabricated design on this machine utilized box sections wherever possible because it was found through experience that this construction lent itself best for our operations. At the same time, it made possible obtaining of the required section modulus for maximum load concentrations with a minimum of weight. It is also possible to obtain much better warpage control with a box-section construction because it tends to equalize the welding on both sides of the neutral axis. TABLE II is a breakdown of the manufacturing cost of the fabricated moldboard.

A-FRAME ASSEMBLY: The A-frame, or push arm, *Fig.* 3, is connected to the center of the moldboard with a universal action type of trunnion. Ends of the side members of the A-frame have trunnion bearings which in turn locate on the trunnion pins of the mounting bracket. In the side members of the A-frame three holes usually are drilled which, when connected with the extension arm, locate the blade in its three operating positions, viz., bulldozing, angling to the left or angling to the right. This A-frame must have the ability to absorb all the load imposed upon it by the moldboard and transfer it into the mounting brackets located on the tractor frame.

Design Employs Truss Principle

The truss principle was maintained in the design of the A-frame and all external loads or forces were applied to the A-frame at the joints so as to cut down flexural stresses in the beam. During operation the A-frame members undergo a reversal of stress from compression to tension depending upon the type of work and the load application points. In addition, the A-frame members must withstand torsional loading. All this was taken into consideration and these members were designed to take maximum compression and torsion under the most adverse loading conditions.

In the new design the A-frame consists of a completely fabricated frame of one-piece construction. Side members are $\frac{3}{8}$-inch plate formed into a box section of two fabricated channel sections welded along the neutral axis. Front members of the A-frame also are $\frac{3}{8}$-inch plate and of similar box-section construction.

Side box-section members of the A-frame contain the holes for location of the extension arm pin. Each of the holes is backed up by a block welded to the inside of the top and bottom member of the box section to increase the bearing area for the pin. These blocks are welded to each half of the box section before it is assembled. The box members are then spotted together and completely welded on a submerged-arc welding machine. The front box section members are handled in a similar manner.

After the side box member is completely welded, it is taken to the drill press and the three extension pin holes are jig drilled through. Front and side box members then are ready to be set up in a locating fixture for welding. The front box member dovetails into the front of the side box member and these members are so located in the fixture that the relationship between them is maintained relative to the three holes of the side member. Side and front members of the A-frame are further tied together by a box-section type of corner gusset. These corner gussets are then located and tacked into position.

Apex of the A-frame consists of a $1\frac{1}{4}$-inch formed top and bottom plate, closed at front and rear with a $\frac{1}{2}$-inch formed reinforcing plate. A $\frac{3}{8}$-inch fillet weld is used to tie these members together and is deposited in one pass with a 5/16 inch rod. Top and bottom plates are drilled before welding and are located over a pin in a welding fixture. A spacer is put between these two plates to guarantee proper relative space for the trunnion bearing. Fixture employed is a rotary type which allows the operator to tack weld on all sides of the A-frame.

After the A-frame has been properly tacked together the rear trunnion bearings are located. These trunnion bearings are $1\frac{1}{4}$-inch bar stock drilled out

Fig. 5—Design of trailbuilder bracket which must withstand severe torsional and bending load components

for the trunnion pin and milled so as to form a removable cap. This cap is held in position with studs and allows the disassembly of the A-frame from the tractor with very little effort. Location of the trunnion bearings is of paramount importance in that a definite relationship must be held between the center pivot hole and the three pin-adjusting holes in each of the side-arm members. If there is too much variation between these three points, difficulty would be experienced in the angling of the blade. The trunnion bearings are located in position as the last item, after all the other parts have been securely tack welded and all distortion has been compensated.

The A-frame then is bolted to a fixture on a weld-

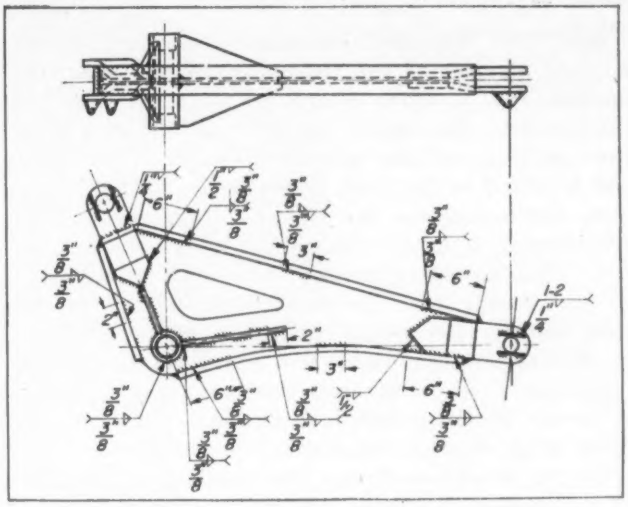


Fig. 6—Lift-arm assembly, of I-beam construction, is subject to reversal of stresses when in operation

TABLE II

Moldboard Assembly—Fabricated Design

MATERIAL AND LABOR COST

Description	Weight of Part (lb)	Material Cost (dollars)	Labor Cost: Machine (dollars)	Labor Cost: Fabricating (dollars)
Adjusting Bracket	98.3	2.949	0.763	1.675
Bracket Bearing	4.6	0.563	1.325	
Trunnion Pin	23.5	1.345	0.875	
Gusset	25.6	0.768		0.356
Brace	25.2	0.756		0.372
Cleat Base	33.6	1.008		0.248
End Plates	38.2	1.146		0.401
End Wear Plate	8.9	0.546		0.180
Pivot Base	21.0	0.630		0.250
Gusset	25.0	0.750		0.435
Top Tunnel	83.5	2.505		0.789
Shim	1.2	0.036		0.053
Spacer	9.0	0.270		0.298
Support	10.6	0.318		0.485
Front Sheet	326.5	16.150		1.450
Reinforcing Plate	6.9	0.207		0.256
Support	8.2	0.246		0.405
Bottom Tunnel	159.2	4.776		0.975
Angle	44.7	1.850		0.485
Reinforcing Bar	5.6	0.168		0.256
Reinforcing Bar	11.4	0.342		0.375
Reinforcing Bar	1.2	0.036		0.157
Reinforcing Bar	1.0	0.030		0.156
Adjustment Ears	125.0	3.750	1.980	2.785
Hook	3.0	0.150		0.100
Total	1100.9	41.255	4.938	12.942

WELDING COST

Description	Footage	Cost (dollars)
1/4-in. Fillet	74′–8″	8.100
5/16-in. Fillet	3′–4″	0.420
3/8-in. Fillet	42′–8″	5.505
1/2-in. Fillet	4′–8″	0.750
1/4-in. Lap	11′–9″	0.725
5/16-in. Lap	12′–0″	0.895
3/8-in. Lap	8′–4″	0.925
Total Cost		$17.320
Complete Fit-up and Spotting Cost		$10.750

COST SUMMARY

Material Cost	41.255
Total Labor Cost	28.180
200 per cent Overhead on Labor	56.360
Welding Cost	17.320
Manufacturing Cost of Moldboard	**$143.115**

ing positioner, being located by the two forward holes of the side arm and the trunnion hole in the center. The positioner allows for all down-hand welding and a procedure has been worked out for staggering the welds so as to compensate for distortion and eliminate extreme stresses. After the A-frame has been welded it is checked for squareness, and in the majority of cases no straightening is necessary.

In employing this type of construction for the A-frame it was possible to eliminate many parts. All of the items which require machining are machined when they are small and easy to handle. Having worked out the procedure, it was possible to obtain high production with a minimum manufacturing and handling cost. Also obtained was an A-frame which, after final welding, was held to tolerances as close as those that could be maintained if the piece were first completely welded and then machined.

EXTENSION ARMS: Third member of the push arm group is the extension arm, *Fig.* 4. This extension arm connects between the end of the moldboard and any one of the three holes in the side arm. Its function is to locate the trailbuilder in one of the three positions, i.e., bulldozing or straight position, trailbuilding or angling either to the right or left positions. The extension arm also controls the tilting of the corners of the moldboard. These arms must be strong enough to withstand a load applied against the moldboard and transfer it to the push arm or A-frame.

During operation these arms can be either under compressive or tensile loading. This was taken into consideration in the cast design and a hat-shaped type of cross section was selected. With this cross section it was possible to obtain a section modulus adequate for the loading and one which was simple enough to adapt itself well to the design.

In the new design, the extension arms are similar in construction, being hat shaped and made of 5/16-inch mild-steel plate. Welded to the front of these extension arms are ears of 1¼-inch plate formed to dovetail into the ears of the moldboard. End tilt adjustment in this construction is accomplished by location of the ears in the proper slots created by the ears in the moldboard and tying both sets of ears together with a curved pin.

On the rear of the extension arms 1-inch thick drilled plates are welded which act as bearings for the extension arm pin. The rear end of the extension arm is further reinforced by gussets similar to that in the cast construction. In both constructions the ears were drilled before welding. The bearing blocks and ears are located in a fixture and then tack welded to the channel section. After this spotting operation, the extension arms are put on a posi-

tioner and four are welded at one time.

Mounting Bracket Assembly: Another highly stressed member of the trailbuilder is the mounting bracket, *Fig.* 5, which mounts to the track frame of the tractor. The upper end of this bracket carries the operating cylinder and lift arms, while the lower end is bolted to the track frame and contains a trunnion which receives the thrust of the push arms, transferring it in turn to the dead axle of the tractor. This mounting bracket must also be designed so that it will clear the grousers of the tractor and then transfer the weight and action of the cylinders to the approximate center line of the track. During operation of the trailbuilder this bracket is subjected to severe torsional and bending loads and therefore must be of rigid and substantial construction.

In the fabricated design this particular piece, due to its complex structure, required quite a bit of study and experimentation before it could be manufactured economically. The final design arrived at consisted of a formed main sheet made up of 5/8-inch material, properly reinforced, clearing the grousers and transferring the cylinder location and operation over the center line of the track. This main sheet was of rectangular appearance, having the material removed from the center for lightness. The removed material then was used to form the saddle to carry the operating cylinder and lift arm at the top of the mounting bracket. The two vertical sides of the mounting bracket were further reinforced with 5/16-inch channel members which were formed to conform to the main sheet, and when welded made two vertical members of a box-section construction.

Lift Arm Assembly: The lift arm on a trailbuilder, *Fig.* 6, is the connecting link between the cylinder and the moldboard. It usually is triangular in shape. The apex of the triangle is pivotally connected to the mounting bracket, the cylinder operating at one corner of the triangle and the lift links connecting to the other corner, actuating the operation of the blade. Generally the lift arm is designed so that a multiplication of the cylinder stroke is obtained which results in faster blade travel and a greater distance of operation of the blade. In a hydraulic machine, this lift arm must undergo a reversal of loading and stresses in the cycle of operation. Parts of this arm which are in tension when the blade is being lifted, are under compression when a down pressure is put on the blade.

In the fabricated design, the mounting frame is constructed so that the lift arm trunnion is located above the tracks. By designing it in this manner a relatively simple type of lift arm resulted. The fabricated lift arm consists of a 1/2-inch thick center web plate reinforced on top and bottom with 3 by 5/8-inch bars to make an I-beam construction. This center web was torch cut to shape and relieved in the center for lightness. A hole was also cut in this web to take the trunnion tube at one end of its triangular shape, and ears were welded at the other two ends. The upper ears connect to the clevis of the cylinder and are formed of 4 by 1-inch bar stock. The two ears at the forward end, which connect to the lift link, are also formed of 4 by 1-inch bar stock. Both of the ears and the trunnion tube were completely machined before being welded to the lift arm. This was done to keep handling and machining costs at a minimum, and also to fit the welding procedure.

Machining Before Welding Effects Savings

In machining these parts when they are small and light, only simple fixtures are necessary, and high production results with a minimum unit cost. This design provides a comparatively light and inexpensive arm without sacrificing strength. After welding, brass bushings were pressed into the trunnion bearings and the only machining required was reaming to insure perfect alignment.

Cylinder Assembly: The cylinder, *Fig.* 7, which is carried in the mounting bracket, supplies the power to the lift arm which in turn actuates the operation of the blade. This cylinder usually is double acting because in the hydraulic machine the prime requisite is to have power up and down on the blade. The oil which actuates the operation of this cylinder is supplied by a pump which can be mounted either in the front or rear of the tractor.

Material selected for this cylinder was seamless steel tubing, and a special machining procedure had to be worked out to co-ordinate with the welding procedures to make this new type cylinder of economical construction. A production line with special tooling equipment was designed so that the cylinder moved from operation to operation. In this line,

(*Concluded on Page* 185)

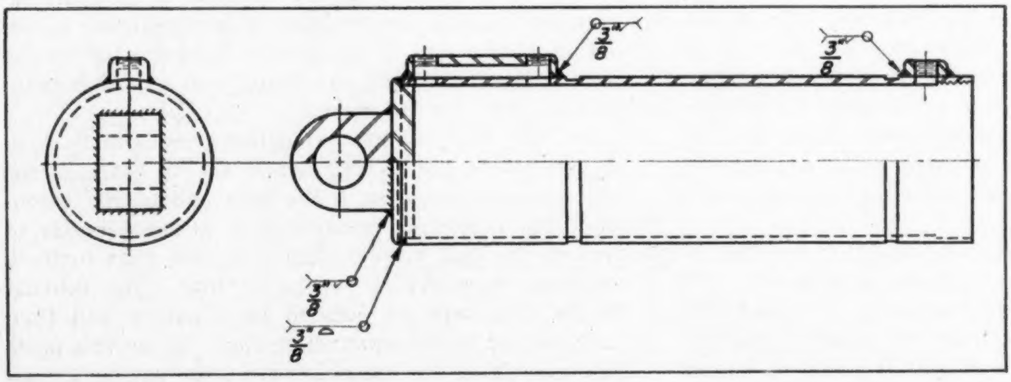


Fig. 7—Hydraulic cylinder for powering blade of trailbuilder is fabricated of seamless steel tubing, bar and plate

Fig. 1—Operator of this grinder views workpiece profile on screen to guide grinding operation

Combination Grinder-Comparator

. . . its design features include two-speed handwheel controls having novel clutching and declutching mechanism

COMBINING the functions of a profile grinder and an optical comparator, the machine pictured in *Fig*. 1, manufactured by Cincinnati Milling and Grinding Machines, Inc., permits continuous, enlarged comparison of work profiles with "master" drawings while the grinding operation is in progress.

The machine comprises two major structural elements; the wheelhead unit carried on a saddle-type slide on the base, and the work-holding and projecting unit carried on a rigid, bridge-type structure which straddles the forward portion of the wheelhead saddle slide.

In the projecting unit, *Fig*. 2, the lamphouse, mounted on the lefthand end of the work-support bridge, produces a brilliant flood of light which is concentrated on the work profile by a pair of condenser lenses. The projection lamp is adjustably mounted in a bracket on top of the lamp housing to permit centering of the filament on the axis of the condenser lenses. The condenser lenses are carried in a barrel-type mount and can be adjusted to focus a parallel beam of light over the work profile. The projection bulb is suspended within a cylindrical sleeve of heat-resistant insulating material and an air space is provided between sleeve and casting. In addition, a small motor-driven blower unit, built into the base of the lamphouse, forces cooling air upward through the sleeve.

Located at the right-hand end of the work-support bridge is the optical projection unit, which provides the enlarged silhouette of the workpiece. This unit

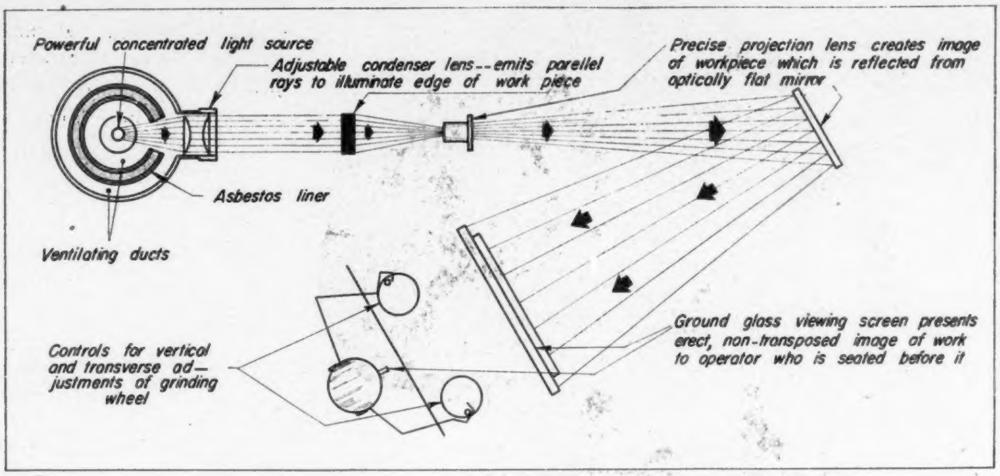
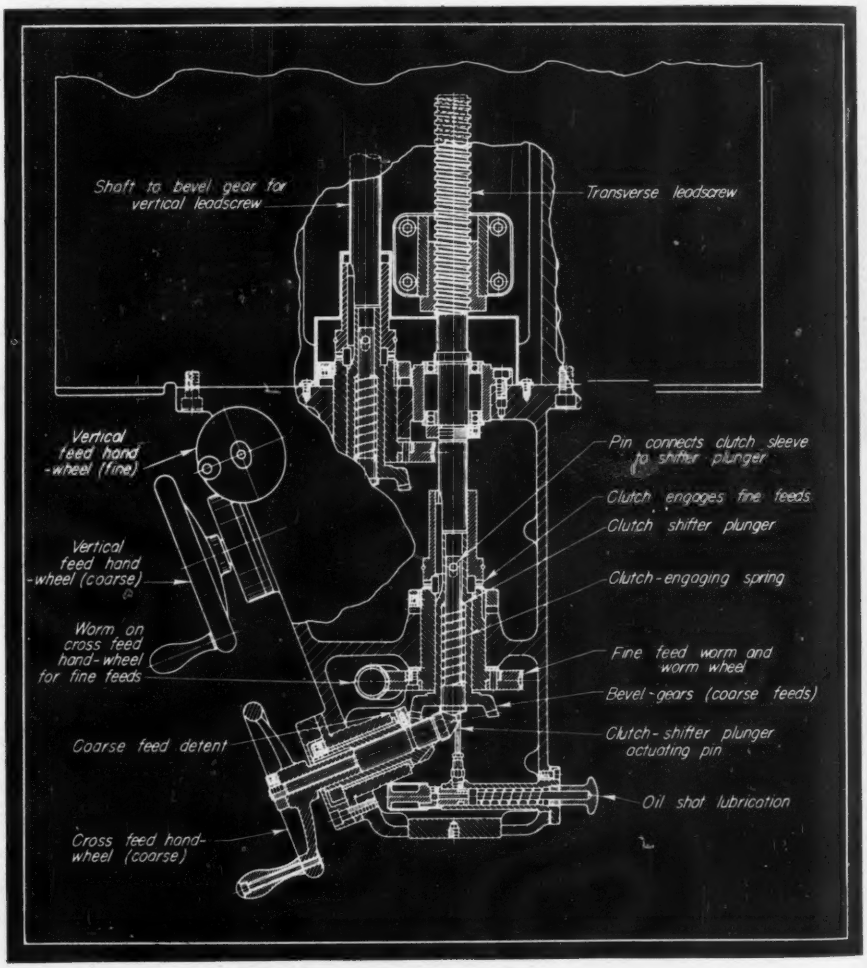


Fig. 2—Left — Schematic illustrates functioning of optical projection unit

Fig. 3 — Below — Handwheel controls of grinder showing how fine-feed declutching mechanism is designed



is completely enclosed within a light-tight hood to maintain as brilliant a screen image as possible without dilution by the normal room lighting. Projection lenses employed are fully corrected to produce a sharp image of the work without fuzziness or color fringe and the screen image is an exact geometrical enlargement of the workpiece.

Lenses are carried in prefocussed, self-aligning mounts to speed and simplify changes of magnification ratio. The image created by the lens is reflected from an optically flat mirror onto the ground-glass viewing screen directly in front of the operator. The silhouette image on the screen corresponds exactly to the work profile as viewed from the operating position, without inversion or transposition, thereby preventing confusion in the positioning of work or grinding wheel. The viewing screen is carried on four antifriction rollers in a ring-type mount to permit angular positioning of the screen and template.

Handwheel controls for the saddle and vertical slides are carried by a bracket which projects from the forward portion of the saddle, in the operating position. Each of the slide movements is controlled by a pair of handwheels, *Fig.* 3, a large one for rough positioning and a small one for fine adjustments. One revolution of the large handwheel shifts the grinding wheel 0.125-inch in relation to the work. The small handwheel, operating through a worm and wheel, provides extremely sensitive positioning of the grinding wheel since the wheel moves only 0.001-inch for each complete revolution of the handwheel.

Inasmuch as the worm and wheel drive for fine feed is self-locking (wheel cannot drive worm), a means must be provided for disengaging the fine feed during coarse feeding. As *Fig.* 3 shows, this is accomplished by incorporating a clutch between the transverse leadscrew and the wormwheel hub. For coarse feeding, the large handwheel and its clutch-shifter plunger is pushed in, pushing the clutch-shifter plunger of the wormwheel unit to disengage the clutch. The detent shown in the handwheel unit keeps the clutch disengaged until the handwheel is pulled outward.

MACHINE *Editorial* DESIGN

Co-ordinated Effort Is Essential

Much more collaboration than in many cases now exists is needed between engineering departments of machine-building companies and other departments including production, sales and research. Engineers responsible for design have, in the past, been "fenced in" to the extent they have had little opportunity for contact with the men and the methods of other divisions of their companies. They have, incidentally, had still less chance to visit the concerns in which the machines they develop have been, or may be, installed.

In many progressive organizations, certainly, committees are established comprising engineers from the various departments involved in the development of each new project. While these function satisfactorily in most instances, individual engineers often need to go beyond the scope of such committees for more intimate data. They should be given much more encouragement in getting together with production men, research engineers, salesmen and others whenever the occasion demands.

Taking as an example the current need for economy in view of soaring prices to illustrate this point, the engineer who is best informed as to how to simplify his design and thus reduce manufacturing costs is he who is ever watchful of up-to-date production methods, utilization of standard parts, analytical selection of materials, and other factors bearing on the cost of the finished machine. Discussion of these points with men in the respective departments concerned cannot fail to enlarge the background upon which the engineer bases his design and from which he may need to present his views at manufacturing committee meetings.

That economical, sound design can best be achieved through co-ordination of effort was amply demonstrated at the recent annual meeting of the Amercian Society of Mechanical Engineers. A panel discussion held at a joint session of the Machine Design and Production Engineering divisions of the society brought together many engineers and served as a forum for presentation of ideas and techniques from both sides. This type of joint meeting should go far toward knitting together the work of engineers in striving for the common goal of production at the lowest possible cost.

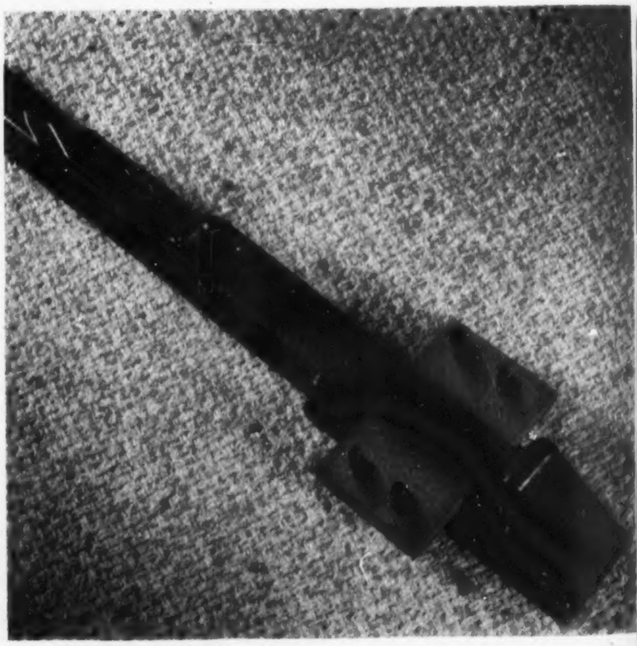
L. E. Jermy

Applications

of engineering parts, materials and processes

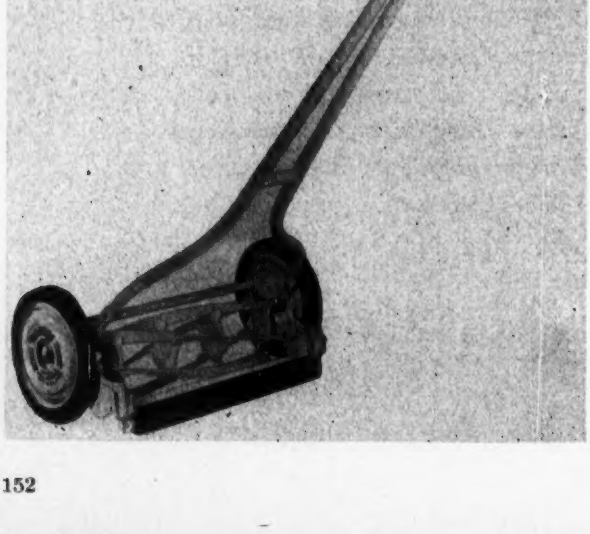
Withstands High Stresses

HIGH-VOLTAGE stresses in the electrical cutout, below, are withstood by a combination part consisting of a Mycalex insert in a phenolic base. Properties of Mycalex (a glass and mica compound) enable the cutout to handle severe electrical and mechanical shock, while the phenolic serves as an inexpensive and sturdy support. Interesting feature of the unit shown is the double use of inserts. Mycalex, itself an insert, has metal studs securely molded in place to receive a contact bar.



Eliminates Fire Hazard

FIRE hazards in the plastics-coating machine, above, have been reduced by the use of the Ionotron static eliminator. Without this device, static electricity, generated by the motion of the paper through the machine, would tend to build up and discharge as a hot spark, igniting the volatile solvents used in the coating compound. The static eliminator contains a radioactive material which emits alpha rays, ionizing the vapor and making it an electrical conductor. Static charges are consequently grounded as they are generated.



Reduces Weight

USE of steel tubing in the Carefree lawnmower, left, has enabled the Reading Hardware Corp. to produce a light, yet strong, machine. Weighing only 37 pounds, the lawnmower has a welded, single-piece frame which forms a rigid unit that cannot loosen under the hardest usage.

Torsional Vibration Frequency

... may be determined quickly to sufficient accuracy for preliminary estimation purposes by use of the charts and formulas presented in this data sheet, which is based on an extensive SAE paper

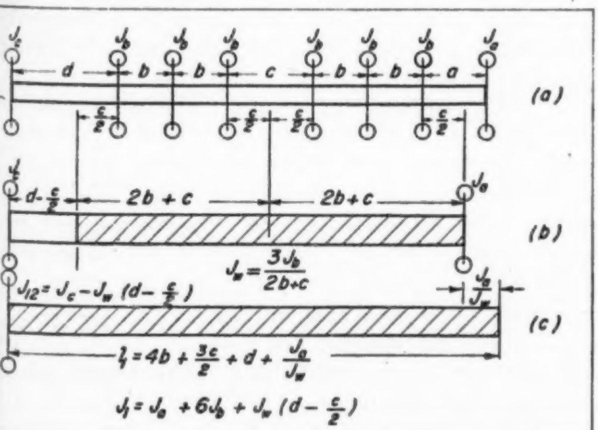
By Frederic P. Porter
Chief Design Analyst
Fairbanks, Morse & Co.
Beloit, Wis.

MATHEMATICAL analysis of torsional vibration of a vibrating system requires, in the first place, the evalution of an equivalent mass-elastic system which will have approximately the same torsional vibration characteristics as the actual system. This is called the "equivalent system," "mass-elastic system," etc. The detailed work for this requires the numerical evaluation of the inertia factors and stiffness factors for all the moving parts of the installation. The final form of the equivalent system depends upon the method to be used in the vibration solution.

One type of equivalent system is obtained by considering the moving masses at each cylinder, flywheel, alternator, and other large compact parts as concentrated masses connected by elastic shafts having no mass. This equivalent arrangement is called the "concentrated-mass system."

Another type of equivalent system is obtained by averaging the various masses distributed along the shafting of the installation into a number of connected uniform shafts having both mass and elasticity. This equivalent arrangement is called the "uniform-shaft system." In the general application of this method certain of the larger masses, such as heavy flywheels, may be concentrated at certain points of the uniform shafts.

Fig. 1—How to convert a concentrated-mass torsional system to an equivalent distributed-mass system



Nomenclature

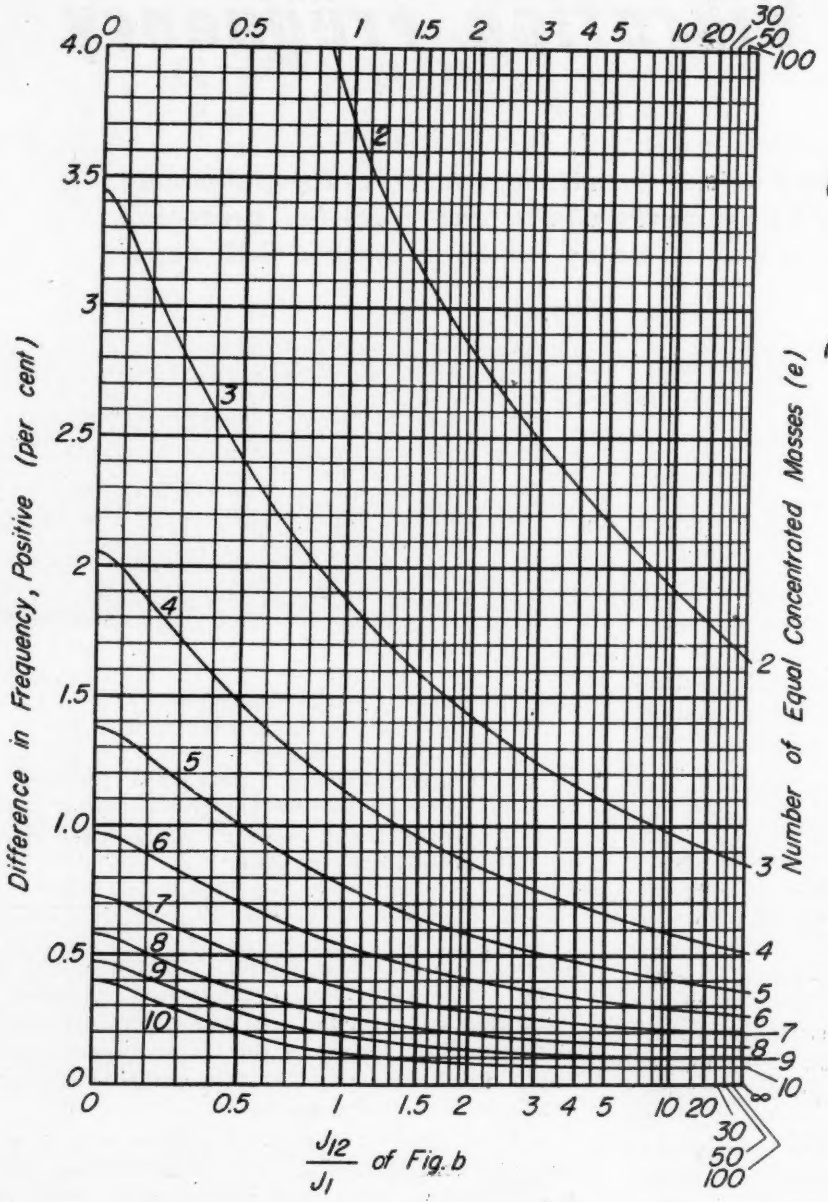
- C = Stiffness factor of shaft (lb-in. per radian) = $\pi d^4 G/32l$ = $1.16 \times 10^6 \times d^4/l$ for steel
- d = Shaft diameter (in.)
- l = Shaft length (in.)
- G = Shearing modulus of elasticity (psi)
- J = Inertia factor or weight polar moment of inertia (lb-in.^{2}) = Wr^2
- W = Weight of rotating mass (lb)
- r = Radius of gyration of rotating mass (in.)
- J_w = Weight polar moment of inertia per unit length of a step with mass (lb-in.^{2} per in.)
- n = Frequency of vibration (vibrations per sec)
- ϕ = 18.3214 $\sqrt{J/C}$ (sec-deg) where J and C refer to a step with mass

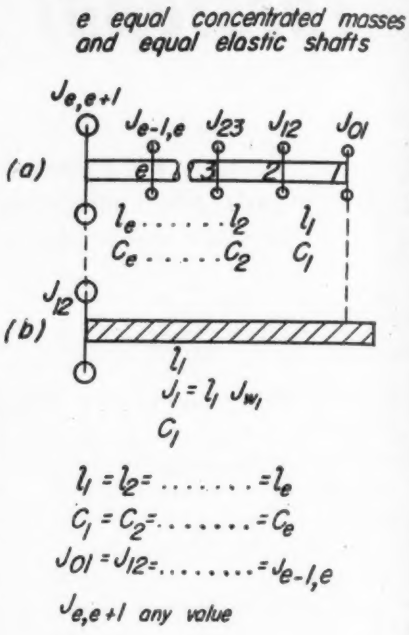
A third type of equivalent system is one which uses any combination of concentrated masses, elastic shafts without mass, and elastic shafts with mass and is called the "combination equivalent system."

Any one of these equivalent systems if determined and treated with sufficient care and detail will give accurate results for the free vibration characteristics provided the inertia factors and stiffness factors are accurately known.

Considerable reduction in the arithmetical work of torsional vibration calculation for multicylinder engines results from considering the crankshaft and masses attached along its length as a uniform shaft with mass and elasticity. How a concentrated mass system for a 6-cylinder engine may be changed to a distributed mass is illustrated in *Fig.* 1. See also the accompanying Nomenclature.

The error resulting from distributing several equispaced, equi-valued concentrated masses with a flywheel at one end is shown in *Fig.* 2. However, these error curves do not apply exactly to an engine with a small number of cylinders since these engines usually have counterweights at each web of the



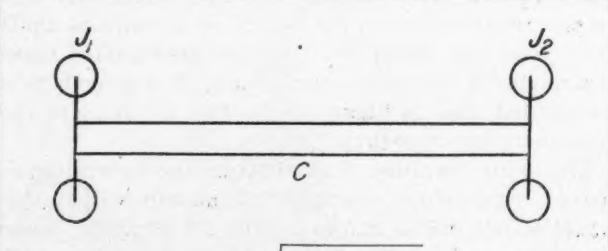


Values for Fig. b in terms of those in Fig. a

Fig. b	Fig. a
J_{w_1}	J_{01} / l_1
l_1	$(e+\frac{1}{2})\ l_1$
J_1	$(e+\frac{1}{2})\ J_{01}$
C_1	$\frac{1}{e+\frac{1}{2}}\ C_1$
J_{12}	$J_{e,e+1} - \frac{J_{01}}{2}$

Fig. 2—Above—Chart shows error in assuming distributed mass (b) instead of concentrated masses (a)

Fig. 3—Left—Formula for natural frequency of a two-mass system with a weightless shaft



$$n=3.127\sqrt{C\left(\frac{1}{J_1}+\frac{1}{J_2}\right)}$$

crankshaft. In this case it can happen that the engine is more accurately represented by a distributed mass than by concentrated masses at the centerline of each cylinder.

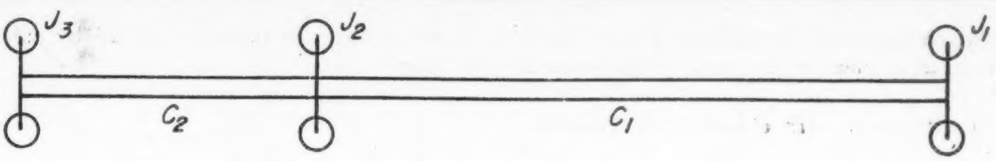
In making solutions for more complicated systems, it often happens that some part of the system has a larger influence in deciding one of the natural frequencies than any of the other parts. This characteristic is usually not difficult to recognize and may be used in preliminary estimates of the frequencies, as follows:

Simple two-mass system, *Fig.* 3

Three-mass system, *Fig.* 4

Uniform heavy shaft, with concentrated masses at each end, *Fig.* 5

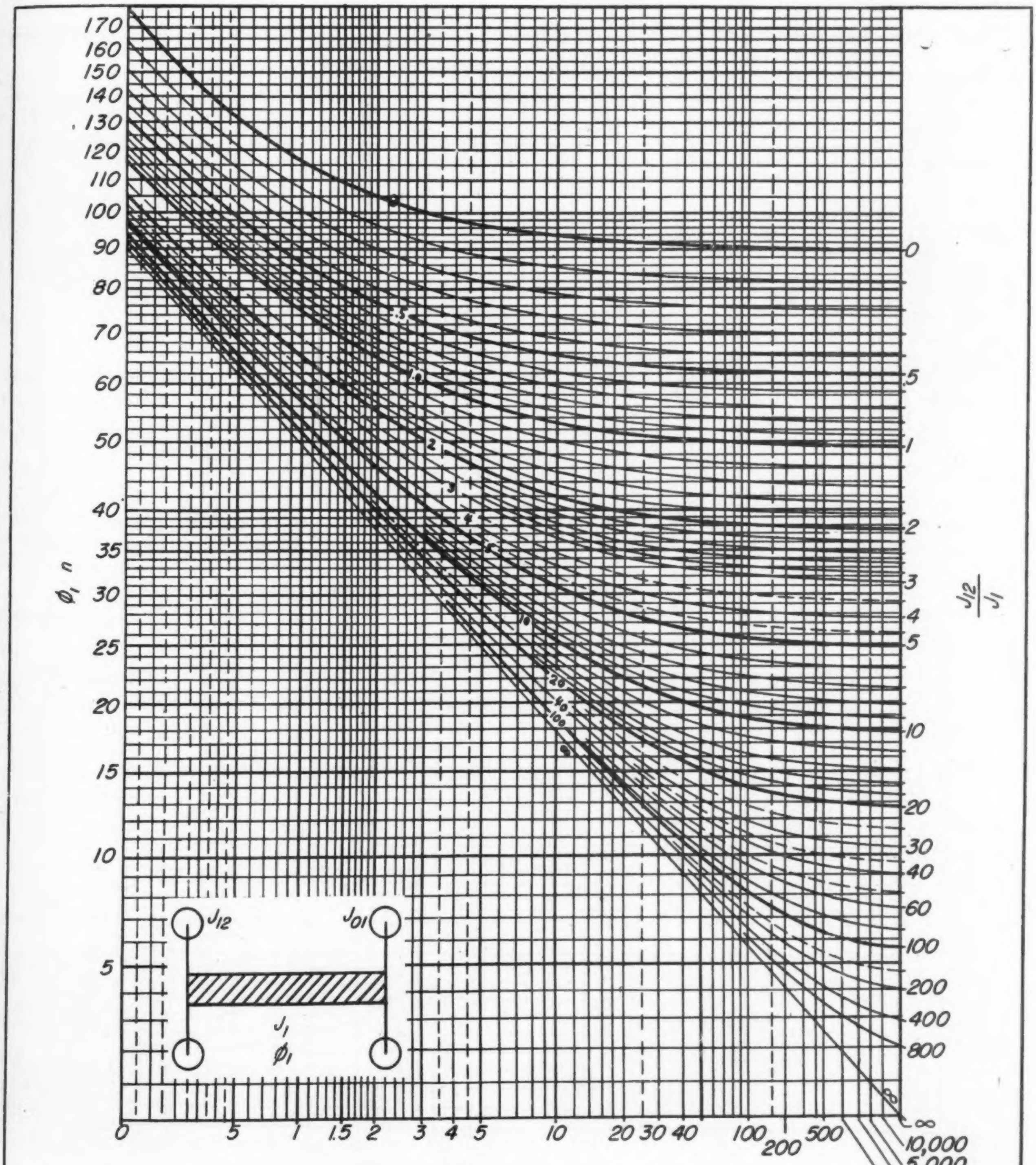
System with two steps, the first with mass and



$$n=3.127\sqrt{\frac{C_1}{2}\left(\frac{1}{J_1}+\frac{1}{J_2}\right)+\frac{C_2}{2}\left(\frac{1}{J_2}+\frac{1}{J_3}\right)\pm\sqrt{\left[\frac{C_1}{2}\left(\frac{1}{J_1}+\frac{1}{J_2}\right)+\frac{C_2}{2}\left(\frac{1}{J_2}+\frac{1}{J_3}\right)\right]^2-C_1C_2\left(\frac{1}{J_1}+\frac{1}{J_2}+\frac{1}{J_3}\right)}}$$

Fig. 4—Above—Formula for the two natural frequencies of a three-mass system with a weightless shaft

Fig. 5—Below—First natural frequency of a distributed mass system with a concentrated mass at each end



elasticity, the second with elasticity without mass, and a concentrated mass at the end of the second step, *Fig.* 6

Uniform heavy shaft with concentrated mass at end, *Fig.* 1*c*:

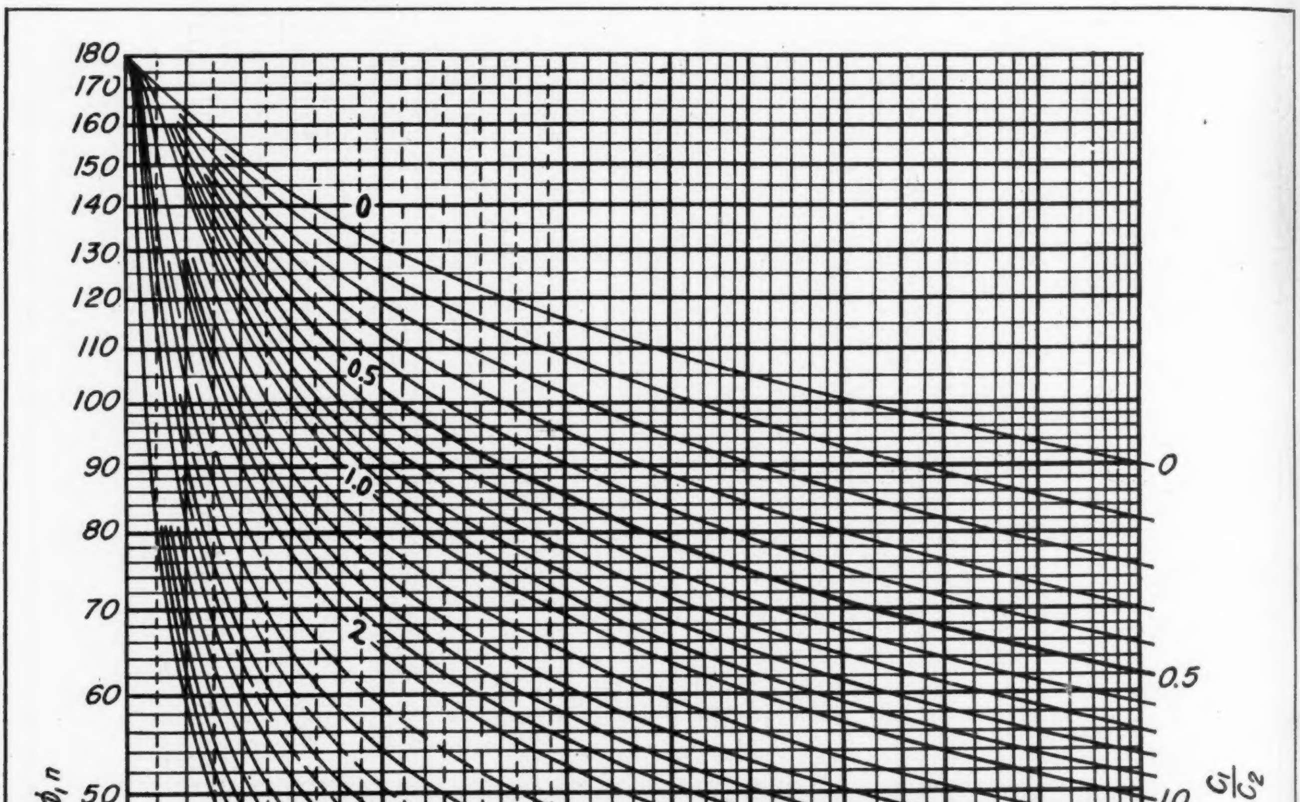
$$n=\frac{\phi n}{\phi}$$

Fig. 6—First natural frequency of a distributed mass connected by a weightless shaft to a concentrated mass

where ϕn (degrees) is found by solving the equation

$$\frac{180}{\pi\phi n}\tan\phi n=-\frac{J_{12}}{J_1}$$

by trial or by referring to tables of the left-hand function. J_{12} and J_1 are defined in *Fig.* 1*c* and ϕ is defined in the Nomenclature. This system also may be regarded as a special case of either *Fig.* 5 or *Fig.* 6.



new parts and materials

To obtain additional information on these new developments see Page 231

Flexible Couplings

Pin-type flexible coupling, newly developed, incorporates the Taper-Lock bushing, up to this time available only on V-belt sheaves. Use of this bushing insures secure fastening to shaft and makes reboring unnecessary. Center disk of the coupling is oak-tanned sole leather, providing flexibility, resiliency and strength. Compact design occupies minimum space on shaft and presents no projecting parts. Center disk can be replaced easily by loosening coupling flanges and sliding them apart on shafts. Units are available in four sizes to suit shafts from ½ to 2½ inches in diameter and are rated to 30 hp. Manufacturer: Dodge Manufacturing Co., Mishwaka, Indiana.

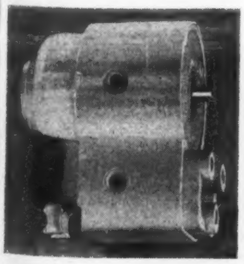
For further information circle MD 1 on card Page 231

High-Temperature Alloy

Cemented-carbide material, known as Grade K138, withstands temperatures that rapidly destroy conventional carbides. It also resists thermal shock, and has a specific gravity about 2/3 that of tungsten carbide. Heated to 2100 F for 48 hours, the material suffers no loss of strength and air cooling from this temperature has no effect other than initial discoloration of the surface. Manufacturer: Kennemetal Inc., Latrobe, Pa.

For further information circle MD 2 on card Page 231

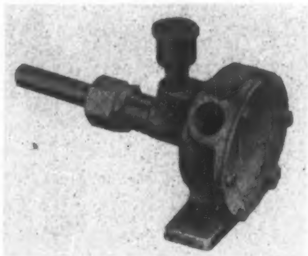
Permanent-Magnet Motor



Miniature permanent-magnet dc motor is designed for use in self-contained units such as portable motion picture cameras or wire recorders. The motor weighs 3¼ oz, measures only 1½ by 1 1/32 by 1½ inches, has a 0.09-inch diameter shaft. Magnet and pole-shoe assembly of the unit is cast-in as an integral part of the die-cast zinc housing. Motor end covers are permanently sealed in to prevent tampering. Revolving at 5300 rpm at no load under 1½-volt (0.23 amp) dc power, the motor can be modified to operate over a considerable range of low voltages. Manufacturer: Essell Corp., 19 Euclid Ave., Newark 5, N. J.

For further information circle MD 3 on card Page 231

Miniature Self-Priming Pump

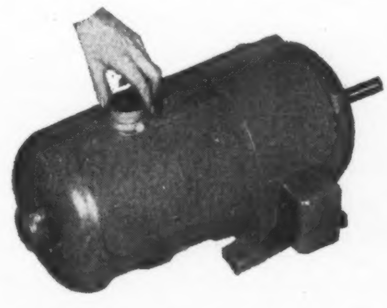


Utilizing a synthetic-rubber impeller, the model A1 pump produces a maximum pressure in excess of 43 psi and will deliver 2.59 gpm against a 10-ft head at 1750 rpm. Smallest unit of the Jabsco line, the pump has port connections for standard ¼-inch pipe with ⅛-inch ports optional. The pump is constructed of bronze with monel shafting, has a total weight of approximately 1¾ lb. Construction of the unit is said to give an especially low friction quality when operating. Unusually long wearing life is claimed since there is no metal-to-metal contact, a factor contributing to smooth, quiet pumping performance. Fluid-lubricated, the pump will not clog and can be operated in either direction with equal efficiency. Manufacturer: Jabsco Pump Co., 2031 N. Lincoln, Burbank, Calif.

For further information circle MD 4 on card Page 231

Adjustable-Speed Electric Motors

Line of Tri-Clad brush-shifting, adjustable-speed induction motors are available in ratings from 3 to 50 hp. The new motors, known as type ACA, feature stepless speed adjustment over a 3:1 ratio by simply turning a dial. Units, with the exception of the starter control, are self contained in a housing only a little larger than that for a constant-speed motor of comparable rating. Motor is rated on a constant-torque basis, carrying full-load torque at rated current and frequency without ex-

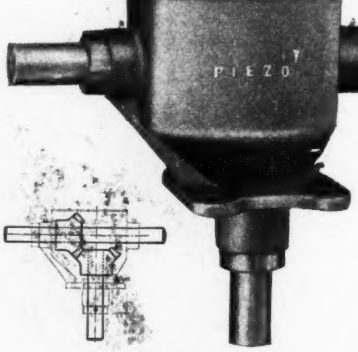


ceeding a temperature rise of 40 C on high speed, or 50 C on low speed. Units are recommended for driving textile machinery, paper machines, wire drawing machines, presses, pumps, etc. Manufacturer: General Electric Co., Schenectady 5, N. Y.

For further information circle MD 5 on card Page 231

Miter-Gear Boxes

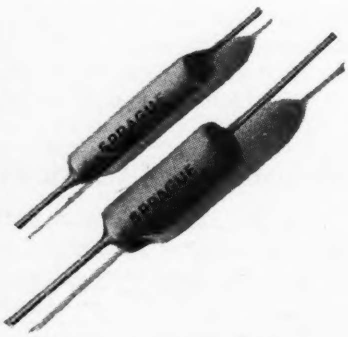
Bevel-gear assembly for transferring shaft motion in mutually-perpendicular shafts, will accommodate shafts of ½ to 1¾-inches diameter, in six sizes. Assembly consists of sturdy housing, on mounting base, equipped with pressure-type lubrication fittings and either sleeve or roller bearings. Unit is equipped with oil seals to prevent leakage. Gears are cut steel, meeting AGMA standards. Manufacturer: Piezo Mfg. Corp., 110 E. 42nd St., New York 17.



For further information circle MD 6 on card Page 231

Miniature Tubular Capacitors

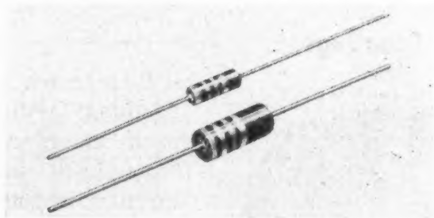
Line of paper-dielectric tubular capacitors are said to be the first tubular types commercially available to operate at 85 C and have ample humidity protection for severe applications. Combining new materials and processes, the types 68P and 69P capacitors are especially suited for conditions where minimum size must be combined with high - temperature tolerance and good performance under high humidity. Type 69P capacitors have side leads to allow wiring across sockets of miniature tubes, while type 68P units have conventional end leads. Manufacturer: Sprague Electric Co., North Adams, Mass.



For further information circle MD 7 on card Page 231

Electrical Resistors

Composition Resistors of the sealed and insulated type are available in ½ and 1-watt sizes with tolerances of ± 5 per cent. Measuring only 9/16-inch in length and 7/32-inch in diameter in the 1-watt size, the new resistors meet all test requirements of Joint Army-Navy Specification JANR-11, including salt-water immersion cycling and high-humidity tests. Known as Little Devil composition resistors, the units are completely sealed and insulated with molded plastic. Leads are soft-copper wire, hardened immediately adjacent to the resistor body, strongly anchored and hot-solder coated. Units are available in



standard RMA values from 10 ohms to 22 meg-ohms. Manufacturer: Ohmite Manufacturing Co., 4980 W. Flournoy St., Chicago 44.

For further information circle MD 8 on card Page 231

Toothed Rubber Belting

Nonslipping drive belt incorporates teeth in the rubber contact surface. The belt is said to be strong, flexible and virtually noiseless in use. It is designed for machinery requiring nonslip operation. Belt is reinforced with steel cables embedded in oil-resisting synthetic rubber. The cables greatly reduce stretch, and are said to eliminate need for take-up devices. In operation the belt makes positive engagement with the pulley at speeds ranging up to 10,000 fpm. Known as the Gilmer Timing belt, the new development will be made in various sizes and is suitable for power transmission and synchronization. Manufacturer: L. H. Gilmer Div., United States Rubber Co., Rockefeller Center, N. Y.

For further information circle MD 9 on card Page 231

Hydraulic Power Unit

Four-speed gearshift drive unit has integrally mounted single-phase motor. The assembly, known as the Type R Drive, is built to drive machinery requiring selective speeds and power in the range ½ to ¾-hp input with speeds of 1200 rpm and 1800 rpm res-

pectively. Gear ratios are 1:1; 1.33:1, 2:1 and 4:1. Units will operate on 115 and 230-volt, 50 and 60-cycle power. Manufacturer: Lima Electric Motor Co., Findlay Road, Lima, Ohio.

For further information circle MD 10 on card Page 231

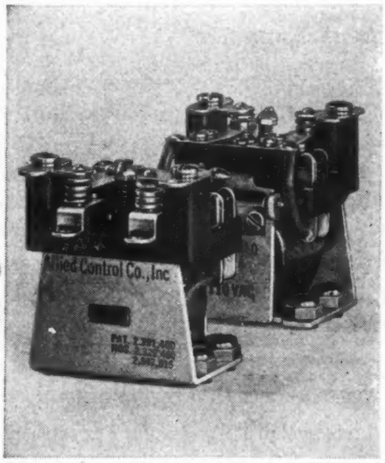
Hydraulic Tube Fitting

Tube fitting employing a hardened spring-steel sleeve is said to seal high fluid pressure, absorb excessive vibration and prevent tube breakage. Fitting consists of three parts, connector body, tightening nut and contractile sleeve which grips the tube with slotted steel fingers when the tightening nut is turned. New fittings are available in sizes from 1/4 to 1 1/2 inches in straight, union, elbow, side tee, tube tee and cross types both male and female. Manufacturer: Flodar Corp., 331 Frankfort Ave., Cleveland.

For further information circle MD 11 on card Page 231

Multi-purpose Power Relay

Designed for convenience of wiring and minimum mounting area, the Allied BO relay is a multipurpose unit designed for either ac or dc operation. Contact arrangement provides double-pole, double-throw

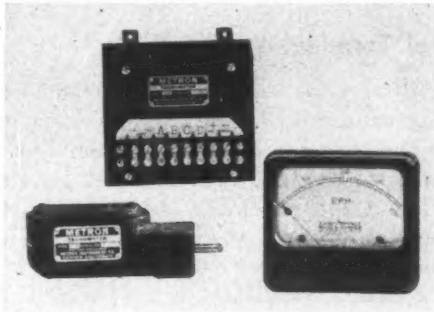


or double-break operation with 1/4-inch contacts rated at 15 amperes. Insulation is molded Bakelite throughout. The relay measures 1 13/32 by 1 5/8 by 1 7/8-inches. Manufacturer: Allied Control Co., 2 East End Ave., New York 21.

For further information circle MD 12 on card Page 231

Instrument-Board Tachometers

Electric tachometer indicators designed to be used with any of the Metron tachometer heads may be used for speed measurement ranging from 1 to 100,000 rpm. The type 42 indicators are sturdily built and equipped with shock mountings to withstand the most severe usage. More than fifty standards speed ranges are available and the dial can be calibrated in any units such as pounds per hour, feet per second, etc. Tachometers consist of three elements: Head, indicator and junction box. Convenient terminals are provided to make the necessary connections. Operating power is obtained from 110-volt 60-cycle line current or available dc supply. One per cent accuracy



is said to be held in spite of age, wear and usage. Manufacturer: Metron Instrument Co., 432 Lincoln St., Denver 9.

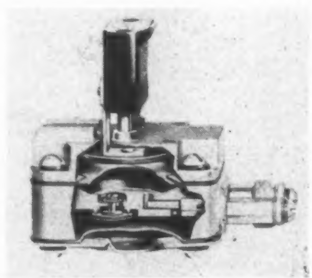
For further information circle MD 13 on card Page 231

Motorized Valve

New, compact motorized valve provides automatic two-position (on-off) operation of valves for steam, gas, oil or water. The unit is operated by an induction-type motor developing a positive drive through a worm and spur-gear reduction. The limit switches are adjustable and designed for trouble-free service. Housing is dust tight and splash proof, with all parts readily accessible. Unit is available in globe-type all-bronze construction, either single or double seated in sizes 1/2 to 2 1/2 inch, with screw-end connections. Butterfly types with iron body and bronze trim are made in sizes from 1 to 4 inches with either screwed or flanged ends. Manufacturer: Automatic Temperature Control Co. Inc., 5412 Pulaski Ave., Philadelphia 44.

For further information circle MD 14 on card Page 231

Pneumatic Timers for Motor Starting



Providing wide-range time-limit motor acceleration unaffected by normal variations in temperature and voltage, new Square D pneumatic timers permit electric motors to be brought up to speed quickly and smoothly. Danger of timing error as the result of friction between surfaces or moving parts, mechanical wear or foreign material on magnet surfaces has been eliminated. In operation.

air is transferred from lower to upper chamber through metering orifice which may be adjusted by means of a control knob. Air returns through valve for instantaneous reset. Timers may be coupled to contactors or actuated by separate magnets. Manufacturer: Square D Co., 4041 North Richards St., Milwaukee 12.

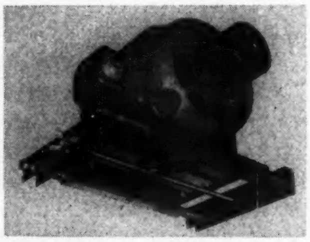
For further information circle MD 15 on card Page 231

Electrical Terminal Strip

Fanning strip for assembly of terminal wires prior to placing on terminal strip will accommodate up to 20 wires and thus greatly reduce assembly time. Strip is designed for use with Jones Barrier terminal strips Nos. 141 and 142 providing 20 and 17 contacts, respectively. Units are sturdily built with cadmium-plated brass terminals riveted on a heavy Bakelite mounting base. Maufacturer: Howard B. Jones Div., Cinch Mfg. Corp., 2460 W. George St., Chicago 18.

For further information circle MD 16 on card Page 231

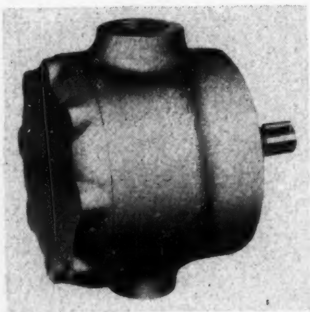
Tension Controlling Motor Base



Capable of being mounted in any position, and having operating features which permit the use of short-center high-ratio belt drives, this motor base is applicable to many designs previously considered outside the scope of tension motor bases. Tension is provided in this new design by two easily adjustable helical springs. Thus, the motor actually floats against correct spring tension so that full and constant power is transmitted to the driven shaft at all times. Unit is available in nine sizes covering motors in ratings from 1/6 to 25 hp. Manufacturer: Automatic Motor Base Co., P. O. Box 2186, Paterson, N. J.

For further information circle MD 17 on card Page 231

Hydraulic Pumps

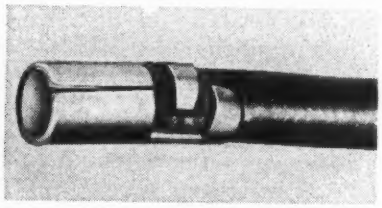


Medium - pressure pumps in a complete line are designed to operate at 1000 psi in continuous duty, or 1500 psi for intermittent service. Soon to be available in capacities from 2 to 75 gpm, the pumps are now manufactured in four sizes: 10, 13½, 20½ and 27½ gpm. Of constant-displacement type, the pump has a hydraulically balanced rotor, thus placing only torque loading on shaft. Features include: Large displacement, new mechanical oil seal design, and ports connecting all pumping areas to assure unrestricted flow. Manufacturer: Sundstrand Machine Tool Co., Rockford, Ill.

For further information circle MD 18 on card Page 231

Solderless Terminals

Solderless ignition-cable terminals having improved snap-on action are said to assure a positive high-efficiency contact with igniter stud. Good electrical contact with the lead is afforded by two insulation-piercing contacts of special design that enter the



cable parallel to the strands of wire, achieving a large area of contact without tending to shear the wire. Manufacturer: Aircraft-Marine Products Inc., 1580 N. 4th St., Harrisburg, Pa.

For further information circle MD 19 on card Page 231

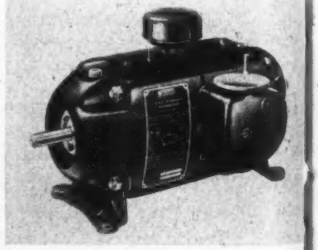
Plastic Clamp Supports

Loop-type cable clamps reduce likelihood of short circuiting and provide a noncorroding, strong support. The plastic clamps are said to save time in mounting and be suitable for supporting conduit, cable, wiring, tubing, etc. They are available in 19

sizes to accommodate tubing from ⅛ to 1½ inches in diameter. Manufacturer: Holub Industries, Inc., Sycamore, Ill.

For further information circle MD 20 on card Page 231

Variable-Speed Drive



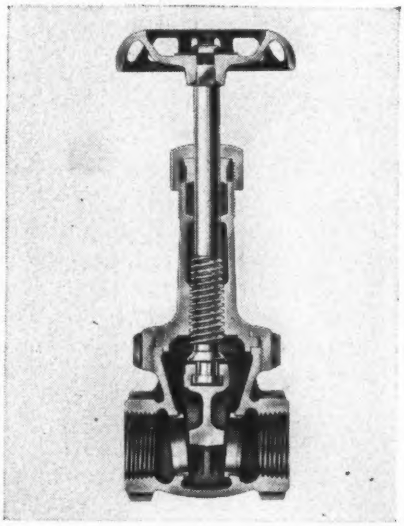
Hydraulic variable speed transmission will provide zero to full speed in either direction of rotation with output torque of 40 lb-in. Unit is available in two models: TR2-HR1-3F13 with handwheel control, and TR2-DR13F13 with servo control. For both units the maximum operating speed

is 1750 rpm and dimensions are virtually identical. Length of the units is 12 19/32 inches over shafts, and width of the slightly wider servo-controlled model is 7 1/16 in. Overall height of both models is 7 3/16 in. Units are self-lubricated, and require no adjustment, they are available in a wide variety of mounting positions. Manufacturer: Vickers, Inc., 1400 Oakman Blvd., Detroit 32.

For further information circle MD 21 on card Page 231

Bronze Gate Valve

Gate valve for 200 psi service is available in sizes from 1/4 to 2 inches. Sturdily built valve has heavy dimensions throughout, has a deep stuffing box to prevent leakage and assure friction-free spindle operation. Principal feature of the new valve is the seat construction. A high-tensile bronze wedge seats against Monel seat rings, having hardness of about 2 1/2 times that of the valve body bronze. The seat

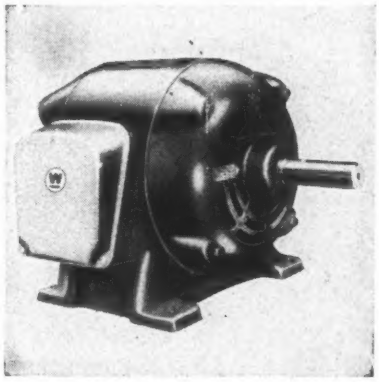


rings are expanded into the body and are equivalent to integral faces. Wear is therefore taken by the easily replaceable wedge. Manufacturer: Jenkins Bros., 80 White St., New York 13.

For further information circle MD 22 on card Page 231

Single-Phase Motors

Capacitor-start induction-run line of single phase motors is available in 1, 1 1/2, 2, 3, 5 and 7 1/2-hp sizes. The new motors embody the all-steel construction of the Life-Line motor, having frame, feet and end brackets of steel. They have self-contained capacitor and transfer switch, the latter disconnecting the starting capacitor as the motor reaches full speed. All openings are in the lower half of the end bells giving full protection against dripping liquids and self-sealed, prelubricated ball bearings will run five years or longer without repacking. Units smaller than two horsepower have built-in manual-reset thermostatic protection against excessive internal temperatures resulting from overloads, too-frequent starting, stoppage of ventilation, etc. Motors are



available for 60-cycle, single-phase, 200-volt service. They will operate successfully at normal frequency and rated load with voltage ten per cent above or below the name plate rating. Operating speed is 1740 rpm. Manufacturer: Westinghouse Electric Corp., 306 Fourth Ave., Pittsburgh 30.

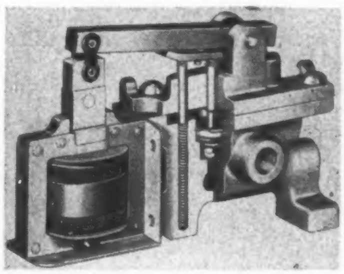
For further information circle MD 23 on card Page 231

Direct-Operating Solenoid Valve

Line of direct and pilot-operated solenoid control valves are made in sizes up to 4 inches. Direct-operated valves range in size from 1/8 to 3/8-inch, while the pilot-operated units are from 1/8 to 4-inches. Of stainless-steel construction, they are suitable for practically all fluid-control up to 3000 psi. Present valves are d-c controlled. Manufacturer: Allied Control Valve Div., Skinner Chuck Co., Norwalk, Conn.

For further information circle MD 24 on card Page 231

Four-Way Pneumatic Valve



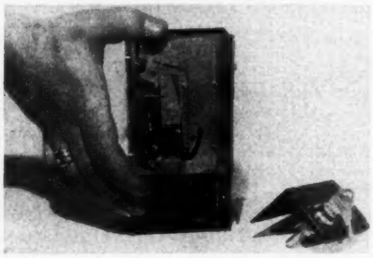
Compact, fast-acting air valve is solenoid operated and spring biased. The unit, which is comprised of stainless-steel balls travelling between opposed closely aligned brass seats, may be operated continuously within a wide speed range, and is engineered to deliver approximately the full volume of the pressure line with minimum pressure loss. Valves in operation have operated up to fifteen million cycles without developing leaks or requiring service. The unit,

measuring 2 by 6 by 4 inches, weighs five pounds with cover and is available in the ¼-inch size only for pressures up to 140 psi. Normally supplied with solenoids to operate on 115-volt, 60-cycle power, the unit has an inrush current of 1.85 amperes and holding current of 0.29 amperes. Models to operate on other a-c voltages are also available. Manufacturer: Crescent Valve Co., 6073 State St., Huntington Park, Calif.

For further information circle MD 25 on card Page 231

Coin-Operated Timer

Dime-operated timer, designed for control of coin-operated equipment, is available in time cycle intervals of 20, 30, or 40 minutes, one or two hours. The 50-D timer is capable of accumulating a reserve of seven coins in its chute for continuous operation over a period of time. Coin box will contain about $15 in dimes. Weighing two pounds, the unit has overall dimensions of 6 7/8 inches high, 3 1/2 inches wide and

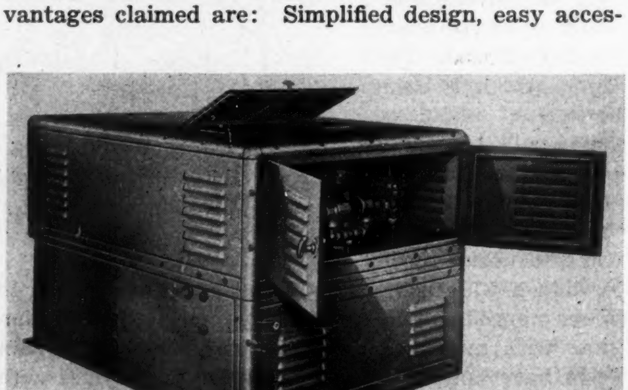


2 1/4 inches deep. It is designed with heavy sheet-metal case, requires 110 volt, 60 cycle power. Manufacturer: American Time Corp., Springfield, Mass.

For further information circle MD 26 on card Page 231

Hydraulic Power Units

Custom-built line of hydraulic power units are designed to incorporate all necessary controls, prime movers and hydraulic accessories within a single "package." Typical unit includes motor, controller, pump, oil reservoir, gages, valves, and filters. Advantages claimed are: Simplified design, easy acces-



sibility, improved appearance, and reduced assembly time. Manufacturer: Vickers Inc., 1430 Oakman Blvd., Detroit 32.

For further information circle MD 27 on card Page 231

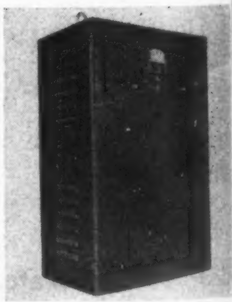
Manually Adjusted Pressure Control

Pressure control, known as model H-701, provides accurately calibrated visible scales for fingertip adjustment in the range 1 to 15 psi. Available for 110

volt, 10 ampere or 220 volt, 5 ampere operation, the unit is a single-pole single or double-throw, normally-open or normally-closed type. It is underwriters approved for ½-hp rating. Outside dimensions are 3⅝ by 3⅝ by 2½ inches. Manufacturer: Thermix Corp., Greenwich, Conn.

For further information circle MD 28 on card Page 231

Electronic Voltage Regulator



Electro-mechanical line of voltage regulators known as type EM has been entirely redesigned to incorporate new electronic and mechanical developments. Modifications include refinement of the control circuit to provide a constant voltage from fluctuating ac power lines. New control-circuit characteristics provide faster detection of line-voltage variations, nonmicrophonic performance, ease of installation and servicing, increased shock resistance, and compactness. All removable components such as the new voltage-sensitive element and the new unbreakable fast-acting relays, are plug-in type. Control is also of plug-in type so that the complete unit may be unplugged from the power elements and replaced by another unit. Manufacturer: Superior Electric Co., 1080 Church St., Bristol, Conn.

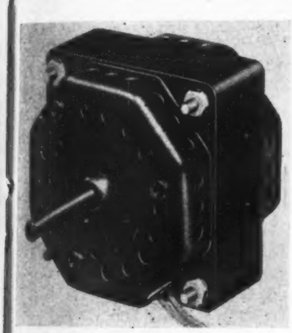
For further information circle MD 29 on card Page 231

Vacuum-Tight Valve

Vacuum valve for use with systems employing mercury pumps has stainless-steel bellows sealing the valve stem from leakage to the atmosphere. Body of the valve is cast medium-carbon steel, cover is high-test iron, and silicone rubber is used for gaskets and valve seat. Valve incorporates many new features including gage connection on inlet and outlet sides of body. Sizes available include, 1, 1½ and 2-inch in screw, flange or welded connection styles, and 3 and 4 inch in flanged and welded styles. Manufacturer: Kinney Mfg. Co., 3529 Washington St., Boston 30.

For further information circle MD 30 on card Page 231

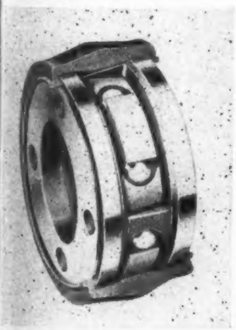
Shaded-Pole Induction Motor



Semi-enclosed four-pole, shaded-pole induction motor measures only 3⅜ inches square and 2⅞ inches long and is rated 1/70-hp at 1600 rpm. The 115-volt motor, which operates on 60-cycle current, is recommended for continuous-duty operation without auxiliary external cooling. Known as the model B, the unit is designed for driving fans, wire recording disks, etc, and may be obtained in larger or smaller horsepower ratings. Manufacturer: Alliance Mfg. Co., Lake Park Blvd., Alliance, Ohio.

For further information circle MD 31 on card Page 231

Ball-Type Flexible Couplings

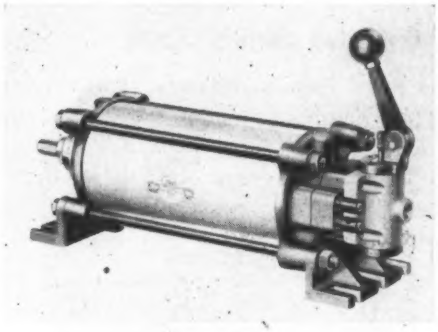


Flexible couplings for speeds to 5000 rpm and loadings to 150 hp will accommodate parallel misalignment up to 1/8-inch, and angularity to 3 degrees. Known as Ballflex, the units utilize ground steel balls rolling between the torque-exchanging members. Only flexible element is the synthetic seal which retains the lubricant and keeps out foreign material, such as dust and dirt. Manufacturer: The Gear Grinding Machine Co., Conant Road and G. T. R. R. Detroit 11.

For further information circle MD 32 on card Page 231

Pneumatic Cylinder

Integral-valve air motor, known as model BM15, will produce a piston thrust 15 times air-line pressure for any line pressure up to 175 psi. Cylinders are available in standard stroke lengths of 1½, 3, 6 and 9 inches with a standard bore of 4½ inches. Units with special strokes are made to order. Valve controls for all motors are integral with the unit. Valve-operating lever is adjustable to work at nearly all angles. Controls to regulate speed of the piston in either or both directions are included. Having forged-



steel heads, heavy brass cylinder, stainless-steel piston rod and corrosion-proofed piston construction, the cylinder is designed for long, trouble-free operation. Unit is available in many styles including pivot mount, remote-control mount, solenoid operated, remote-control manually operated and standard foot-mounted units. Manufacturer: The Bellows Co., 222 W. Market St., Akron, Ohio.

For further information circle MD 33 on card Page 231

New Synthetic Rubbers

Two Hycar rubbers, now available, have been developed for easy processing. Hycar OR-25-EP and Hycar OR-25-NS share qualities of easy mixing, good extrusion and good flow. Both have the property of good "building tack" to facilitate their use in laminated products. The two materials differ in that the type NS has a nonstaining antioxidant, making it suitable for fabrication of light-colored parts where freedom from staining and discoloration are an important factor. Manufacturer: B. F. Goodrich Chemical Co., Rose Bldg. Cleveland 15.

For further information circle MD 34 on card Page 231

Neon Glow-Lamp Indicators

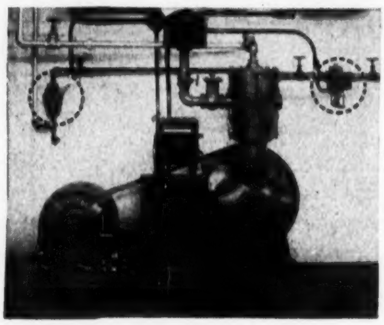
Indicating lamp of the neon type is available in diversified models to met a wide range of requirements in mounting, visibility, appearance and light

brilliance. The units consist of a heat-resisting outer housing which is provided with suitable mounting bracket, and proper resistance for operating on voltages from 70 upward, both ac and dc. Lights are provided with or without shielded lead wires. Manufacturer: American Electronics Corp., 226 N. 4th St., Columbus, Ohio.

For further information circle MD 35 on card Page 231

Solenoid-Operated Control Valves

Solenoid valve handles fluids at temperatures up to 400 F and pressures to 150 psi. It is a direct-operated type with no auxiliary pistons. The unit is a



heavy-duty type, developed to make quick opening solenoid valves available for a wide range of services. Illustration shows valves used to control flow of cooling water on air compressor. Manufacturer: Johnson Corp., Three Rivers, Mich.

For further information circle MD 36 on card Page 231

Variable-Speed Motor Drive

Series of variable-speed drives designated Size 23 are available in a wide variety of assemblies to fit many applications. Units include horizontal frame with shaft left or right, upright frame with shaft high or low, and built-in speed reducers with single

or double reduction. Speed variations of 2:1 to 7:1 are obtainable in units rated up to 5-hp output. Features of the new reducer include: Smaller size, longer belt life, more convenient positioning of the speed-adjusting handwheel and provision for easier change of belts. Manufacturer: U. S. Electrical Motors, Inc., Milford, Conn.

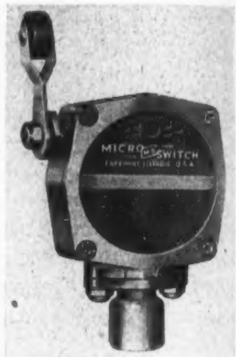
For further information circle MD 37 on card Page 231

Pressure Transmitters

Pressure data transmitters having high accuracy and operating on a large a-c or d-c signal are designed for use with gases and liquids such as air, oil, gasoline and hydraulic fluids. Designated as series-4600, these transmitters combine controlled-action bellows or bourdon tubes and Microtorque potentiometers and can be used with pressures to 500 psi. Accuracy is within 2 per cent. Approximate weight is 1.3 pounds and overall length under 6 inches. Manufacturer: G. M. Giannini & Co., Inc., 285 W. Colorado St., Pasadena 1, Calif.

For further information circle MD 38 on card Page 231

Rotary Actuators

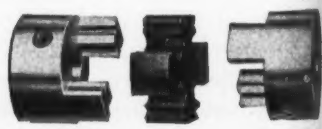


Industrial limit switch housing has actuator consisting of a rotary-action lever arm with roller, the arm being adjustable through 360 degrees in either direction. The unit is operated by a movement differential of one degree or 0.027-inch at the roller and overtravel may be as great as 55 degrees. The LMR5 housing when used with basic BZ-2R-A2 switch is operated by a force of from 4 to 12 oz, with a pretravel of 10 to 30 degrees. It is a sealed unit providing protection against dirt, dust, moisture and occasional liquid splashing. Conduit fitting has standard ½-in.-14 NPSM female threads. Manufacturer: Micro Switch Division of First Industrial Corp., Freeport, Ill.

For further information circle MD 39 on card Page 231

Flexible Coupling

Incorporating a novel tongue and groove feature, the new Climax type C flexible couplings consist of only three parts: Two metal hubs and load insert of rubber. Use of tongue and groove positively locks insert in, preventing extrusion between the jaws. Designed with a large



area of contact between jaws and insert, the new units have great capacity and a maximum of resiliency in shock absorption. Couplings are available in six sizes, for shafts from ½ to 1⅝ inches in diameter, and have power capacities from 1/6 to 12 hp at 1750 rpm. They

are recommended for light-duty electric-motor drives such as on pumps, fans, blowers and small machines. Manufacturer: The Climax Flexible Coupling Co., 363 E. 140th St., Cleveland 10.

For further information circle MD 40 on card Page 231

Flexible Metal Tubing

Flexible tubing for operation at temperatures up to 1700 F has inner core and braid of Inconel. The tubing is recommended for extreme service applications where Monel or brass would be affected. Typical uses include engine exhaust tubing, aircraft fuel and oil lines, food handling and chemical handling. Manufacturer: Titeflex, Inc., 532 Frelinghuysen Ave., Newark 5, New Jersey.

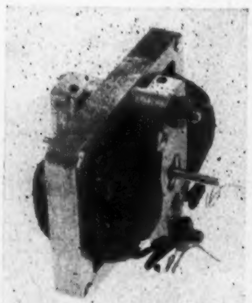
For further information circle MD 41 on card Page 231

Immersed-Type Coolant Pump

Motor-driven coolant pump is designed to handle liquids containing reasonable amounts of grit and abrasives. There are no metal to metal contacts within the submerged portion of the pump. Unit incorporates totally enclosed electric motor, with built-in conduit box and large, sealed prelubricated ball bearings. Twin inlets within the pump are arranged to provide hydrodynamic balance, thus eliminating end thrust on the shaft. Capacity of the pump is 50 gpm with a 26-ft head when powered by a ½-hp motor. When driven by a ¾-hp motor, capacity is 60 gpm. Manufacturer: Ruthman Machinery Co., Cincinnati 2.

For further information circle MD 42 on card Page 231

Synchronous Instrument Motor

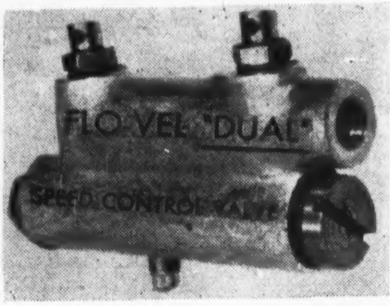


High-starting-torque instrument motor operates at a synchronous speed of 1800 rpm, and is available to suit any normal voltage and frequency. The new B-30-CS motor is similar to the Gleason Avery B-30-A, non-synchronous motor in general appearance, differing mainly in having twice the number of laminations and larger coils. As a result the starting torque available is greater, being 1½ oz-in. The unit is recommended for powering timing instruments, recording devices, dictating machines and other instruments requiring high starting torque and constant speed. Manufacturer: Gleason-Avery, Inc., 27 Clark St., Auburn, N. Y.

For further information circle MD-43 on card Page 231

Hydro-Pneumatic Speed-Control Valve

Cylinder-control valve for use with either hydraulic or pneumatic systems provides high-speed travel for any desired fraction of the stroke, plus precision speed control of the balance of the stroke. Available in models for use with any two-way cylinder, the valves are made in two sizes. Small size handles ⅛ to ⅜-inch pipe, while the larger units are designed



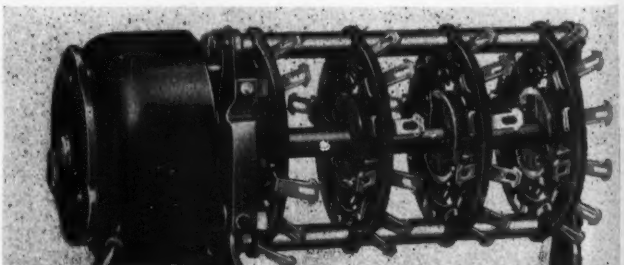
for ½ and ¾-inch pipe. Known as Flo-Vel Dual, the units function equally well with manual or solenoid-operated four-way valves. Manufacturer: The Flo-Vel Co., 413 North St., Sturgis, Mich.

For further information circle MD 44 on card Page 231

Solenoid-Operated Selector Switch

Rotary-Solenoid actuated circuit selector switch is suitable for remote operation of multiple circuits. Solenoid is controlled by a master switch and by a commutating switch synchronized to close at the start of the energized stroke and open at the end of stroke. Solenoid return is powered by a spring. "Homing" contacts in one of the switch sections can be arranged to stop the rotation at desired positions, and other contacts provide for indication of circuits selected. This Ledex unit is available in various values of stroke up to and including 95 degrees. Operation is on direct current. Manufacturer: G. H. LeLand, Inc., 137 Webster St., Dayton 2, Ohio.

For further information circle MD 45 on card Page 231

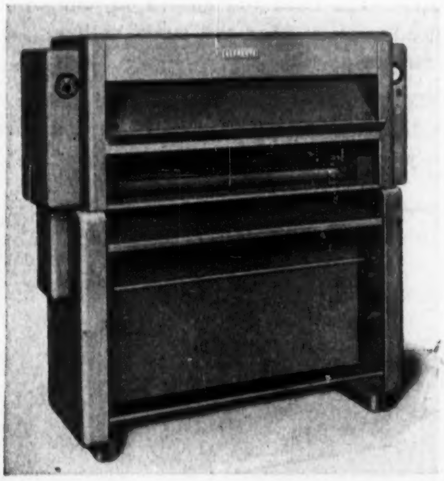


engineering dept equipment

To obtain additional information on this new equipment see Page 231

Ammonia-Type Print Developer

Developing machine for use in processing ammonia-type whiteprints incorporates a number of new features. Perforated steel rollers are used to apply the



ammonia vapor, the rollers revolving at the same rate as the paper to minimize static electricity and sticking. Greater exposure results which permits faster developing speed and shorter distance of travel. This Revolute A Finisher is equipped with a diaphragm pump which meters the ammonia, drawing the fluid from the original shipping container located in the base of the machine. Controls are furnished for providing proper temperature and supply of ammonia vapors. Construction is cast aluminum finished in a gray wrinkle lacquer. Machine develops prints at 15 feet per minute, but units with higher operating speeds are available. Manufacturer: Paragon-Revolute Corp., 77 South Ave., Rochester 4, N. Y.

For further information circle MD 46 on card Page 231

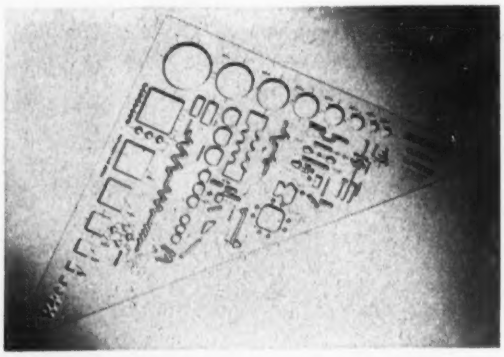
Fluorescent Lighting Fixture

Semidirect lighting unit known as the Grenadier has been modified to have increased shielding both parallel and perpendicular to the lamp tube. Easier maintenance is provided by removable side panels and louvers. Downward component of illumination may be increased by accessory opaque reflectors or by slotted reflectors placed immediately above the lamps. Other improvements include heavier-gage metal, and a new finish. Manufacturer: F. W. Wakefield Brass Co., Vermilion, Ohio.

For further information circle MD 47 on card Page 231

Drafting Template

Plastic drafting template No. 175 simplifies the accurate reproduction of electrical symbols for the light and controls field as shown in the ASA manual, and other accepted symbols. Templates are high-quality



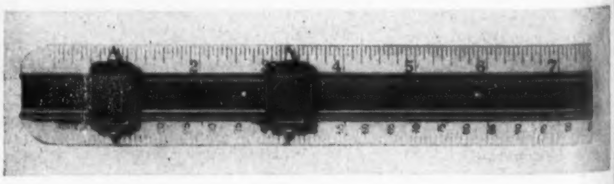
plastics sheet 0.080-inch thick, with inner contours beveled. Printing is between laminated sheets and cannot wear off. Manufacturer: RapiDesign, Inc., P. O. Box 592, Glendale, Calif.

For further information circle MD 48 on card Page 231

Rule with Indicator

Drafting instrument consists of standard rule with two moveable point indicators, permitting user to mark points for frequent use. Device consists of wooden rule with metal rail on which steel indicator fingers slide. Fingers extend down to edge of rule, clearly indicating chosen point. Manufacturer: Chowns Mfg. & Design Co., 16535 Manchester St., East Detroit, Mich.

For further information circle MD 49 on card Page 231



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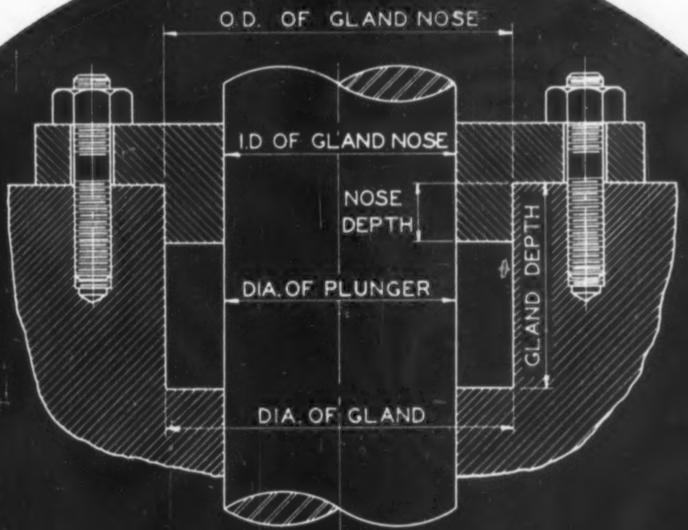

PRESSURE MAX.______ MIN.______

TEMPERATURE MAX.______ MIN.______

MEDIUM______

MOVEMENT • ROTARY RECIPROCATING

SPEED______ PER MINUTE

O.D. OF GLAND NOSE______

I.D. OF GLAND NOSE______

DIA. OF GLAND______

DIA. OF PLUNGER______

GLAND DEPTH______

NOSE DEPTH______

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Noteworthy Patents

PRECISE CONTROL of injection timing relative to engine speed is accomplished by an unusually simple mechanism employed in a fuel pump assigned to International Harvester Co. by Alexander Dreisin. Patented under number 2,417,915 the pump automatically regulates the timing of injections according to the speed of the direct engine drive by virtue of action of a constrained ball race. Inertia force component generated by the whirling balls moves the control cylinder of the pump to advance or retard the registry of the metering parts. Control of the rotational position of the pump plunger affords additional control of fuel quantity injected.

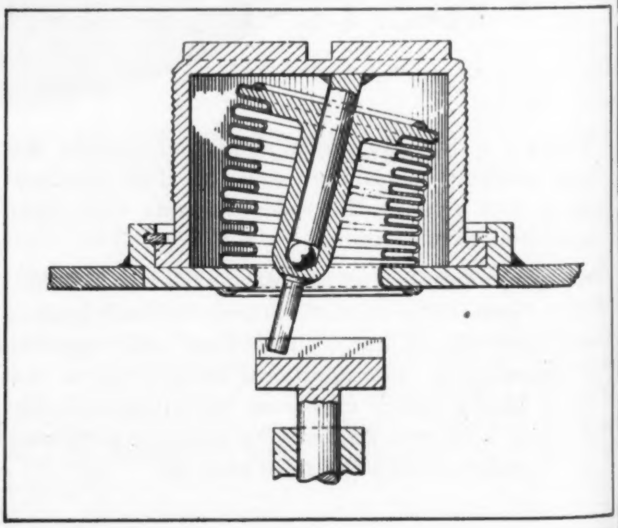
IMPROVED PIVOTED-PAD sleeve bearing for high-speed shafts is covered in patent 2,421,668 assigned to General Electric Co. by David H. Ware. Bearing is provided with two resilient support sections which extend approximately one-third the circumference in each direction from a split opening. Relief or undercut at the attached end and tapering of the resilient section achieves increased flexibility to create pivoted-pad action. Lubricant wedge spaces provide a continuous film of oil which accurately centers the shaft during rotation and assures a minimum of bearing friction.

COMBINATION UNIT one-way clutch and anti-friction bearing utilizes a single cage to space the balls and gripping members of the clutch. Normal rotation of the bearing cage in one direction keeps the grippers free while opposite rotation actuates the gripper to engage the bearing races and lock them with a maximum of free travel. Covered under patent 2,423,178 assigned to Adiel Y. Dodge, Rockford, Ill.

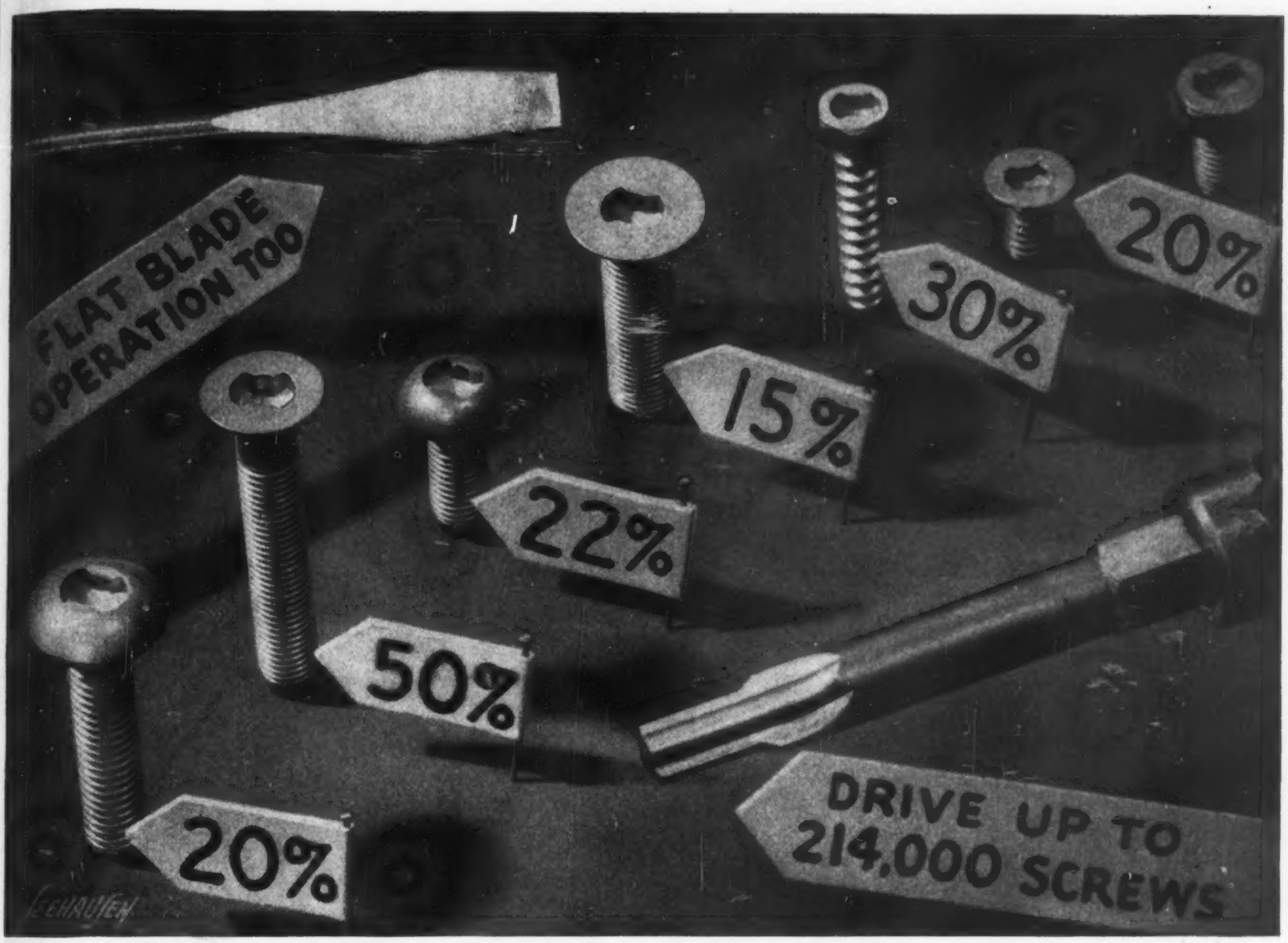
COMBINATION SEALING effect is obtained with a unique shaft seal assigned to General Electric Co. by Robert W. Groot. Detailed under patent 2,418,707, the dish-shaped seal is formed from a flexible material in such a manner that it makes a positive sealing engagement with the machine housing when the shaft is at rest. Under rotating conditions, however, centrifugal force deflects the seal outward away from the housing to effect a slinger type fluid seal which also prevents passage of the fluid along the shaft. Friction normally present in most shaft seals is minimized to a great extent.

EXTRUSION of O-ring seals, when employed at pressures in the range of 3000 psi, is prevented by a novel seal guard ring outlined in patent 2,420,104 assigned to The Maytag Co. by Thomas R. Smith. Triangular in shape, the rings are made of hard metal, split to allow insertion, and with a diameter of such size as to fit the cylinder bore tightly. One used at each edge of the O-ring groove creates a complete walled-in effect to hold the ring under high pressure.

TRANSMISSION OF MOTION from an external source into a sealed mechanism is accomplished without impairing the seal by a device shown in patent 2,419,074. In order to preserve the seal, a flexible bellows is employed having one end free for movement sealed to the lever and the other end fixed so as to seal the interior of the mechanism. Arranged



to operate within the hollow lever is an operating rod fixed at an angle to a swiveling cover piece. Movement of the cover piece actuates the bellows lever to effect a crank motion. Patent assigned to Kearfott Co. Inc., by F. O. Herbert, Jr.



FLAT BLADE OPERATION TOO
20%
30%
15%
22%
50%
20%
DRIVE UP TO 214,000 SCREWS

MEN... of machines

John C. Widman

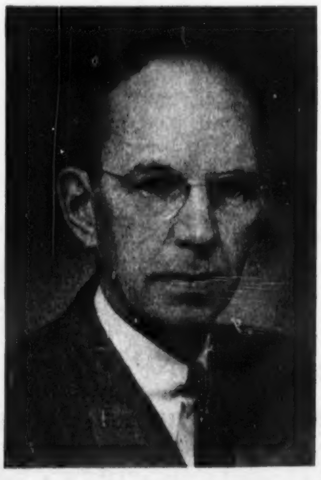
Ralph H. Isbrandt

JOHN C. WIDMAN in his new position with Kaiser-Frazer Corp. as chief body engineer will direct development work on the current and future body designs. Well known in the field of automobile design, Mr. Widman for the past two years was in charge of the body research department of Ford Motor Co. His early experience in the automotive field dates back to 1930 when he joined Murray Corp. of America. He remained with this organization until 1945. Until the start of World War II he was a project engineer in the die model, sample body and other divisions of the company. During the war, however, he served as chief production engineer and later as superintendent of the engineering and quality division of the Murray Corp. Mr. Widman is a graduate of the University of Michigan, and is a member of SAE and the American Society of Body Engineers.

RALPH H. ISBRANDT, until recently vice president and general manager of the Firestone Aircraft Co., has joined Kaiser-Frazer Corp.—as chief chassis engineer. His broad automotive background covers all phases of chassis engineering. He started with the A. O. Smith Co., and later served as a frame specialist and design engineer with Willys-Overland Motors, the Buick division, Nash-Kelvinator Corp. and the Firestone Tire and Rubber Co. where he specialized in developing uses of rubber in chassis construction. During World War II he served with Firestone's aircraft construction interests in Pennsylvania.

JOHN SEAGREN, returning to Atlas Imperial Diesel Engine Co. as chief engineer, will be benefited by his previous knowledge of the company's line and his outstanding achievements in diesel design and refinements. Mr. Seagren left Atlas in 1944 where he had been chief engineer, to accept an appointment as chief engineer of American Locomotive Co., from which position he recently resigned to return to Atlas. His connection with Atlas goes back to 1931 at which time he joined the company and was placed in charge of diesel research and development. He remained in this position until his appointment in 1936 as chief engineer. Born and educated in Sweden, Mr. Seagren served his apprenticeship in the diesel development department of Fairbanks, Morse & Co. prior to joining Atlas.

WARREN R. TUCKER, formerly vice president of Commonwealth Engineering Co., has been elected to fill the newly created post of vice president in charge of engineering research of Hydraulic Press Mfg. Co. Born in England in 1908, Mr. Tucker



John Seagren

Warren R. Tucker

ELECTROLUX saves thousands of dollars per month by using J&L ELECTRICWELD TUBING...

J&L STEEL

...sets new production records, makes better cleaners at lower cost

Electrolux Corporation, a true leader in its field, guards its product reputation carefully. The market that Electrolux has built for its vacuum cleaner in more than twenty years rests solidly on a foundation of efficiency, quality and dependability.

At one time, Electrolux inspectors rejected as much as 35% of the tubes needed for the Electrolux cleaner hose assembly because they did not measure up to the maker's inspection standards. This constant rejection cut production of the more than one million Electrolux cleaners needed yearly to meet tremendous demands.

When a J&L Engineer went to work on the problem he learned the reasons for these rejections. Lengths of tubing which serve as *wands* or handles, nozzles and hose fittings had to be expanded at one end, swaged at the other, bent, rolled and slotted to fit together perfectly. Then, all pieces must first be copper plated, nickel plated and finally chrome plated to give a lustrous, long-lasting finish.

To pass through all these operations and come out as a perfectly finished product, tubing should have uniformity of cross section and thickness, high surface finish, ductility, and be easy to fabricate.

A test run in the Electrolux plant *proved* J&L Electricweld tubing superior on every count! It fabricated without weld failures and took plating excellently. Rejection of tube parts dropped to almost nothing. Production was *stepped up*, costs *reduced*, thousands of feet of tubing saved each month. Electrolux quality was maintained with a saving in time, money and materials—all important to the Electrolux policy of maintaining its prewar price and service to its several million customers.

If your profits are being reduced and deliveries delayed by production losses—if mounting rejections of fabricated tubing are a problem in your quality control—then, by all means, get the facts on J&L Electricweld tubing *now!* Call in a Sales Engineer from the nearest J&L office or write direct for your free copy of the booklet "*J&L Electricweld Tubing.*" The coupon below is for your convenience.

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received his early training at Dulwich College, and later in this country attended University of Chicago and the Lewis Institute where he received his mechanical engineering degree in 1933. Joining the Hydraulic Press Mfg. Co., he successively became field engineer, development engineer and director of research and development during the period from 1933 to 1945. He has been a member of the Board of Directors since 1933. In 1945 he joined Commonwealth Engineering as projects supervisor and at the end of that year was elected vice president for operations and also a member of the board of directors. In November, 1947 he returned to Hydraulic Press Mfg. Co. as vice president in charge of engineering and research. Mr. Tucker has nearly one hundred patents to his credit in applied hydraulics and equipment for operation and control of hydraulic systems. His experience includes the development and design of the following: Equipment for the production of parts from plastics, dies and molds for plastics and metal powders, operating and control mechanisms for high-pressure hydraulic equipment, continuous self-cleaning filtration equipment and auxiliary equipment, and the investigation of flow of fluids under extreme pressures.

♦

WILLIAM RODDER has been promoted to vice president in charge of engineering of The Aetna-Standard Engineering Co. Mr. Rodder has been in the company's engineering department for eighteen years, and has been chief engineer for the past nine years.

♦

HENRY N. MULLER JR., former manager of graduate student training at Westinghouse Electric Co., has been named manager of the entire educational department. GUY KLEIS succeeds Mr. Muller as manager. Formerly he had served as supervisor of engineering training.

♦

JOSEPH MODROVSKY has joined the faculty of Polytechnic Institute of Brooklyn as assistant professor in the department of mechanical engineering. He formerly had been a design engineer with Wright Aeronautical Corp.

♦

A. F. FISHER, vice president in charge of engineering and manufacturing, Telechron Inc., has been named executive vice president of the company.

♦

THOMAS L. HALLENBECK, research engineer for Baker Brothers Inc., has been promoted to vice president and director of engineering.

♦

DOUGLAS M. MCBEAN, widely known consulting engineer and inventor, recently was elected to the board of trustees of Clarkson College of Technology.

♦

DR. R. C. MASON succeeds GAYLORD W. PENNEY as manager of the electro-physics department of Westinghouse Research Laboratory. Dr. Mason had been on leave of absence at the atomic energy project at Oak Ridge, Tenn., for the past year. Mr. Penney was recently appointed Westinghouse Professor of Electrical Engineering at the Carnegie Institute of Technology.

♦

DR. BRUCE S. OLD will serve the Atomic Energy Commission on a part-time leave of absence from his present connection, Arthur D. Little Inc. In the capacity of chief metallurgist of the division of research of the Atomic Energy Commission, Mr. Old will assist in organizing the work of the commission.

♦

ALBERT MUSSCHOOT, who has been connected with the general engineering staff of Link-Belt Co. at Philadelphia, has been appointed assistant to the chief engineer with headquarters at the company's general office.

♦

RAYMOND F. PFAUTSCH succeeds A. G. SUTCLIFFE, retired, as chief engineer of the Ilg Electric Ventilating Co. Since 1940 Mr. Pfautsch has been assistant chief engineer. Mr. Sutcliffe has been with the company for thirty-seven years, being chief engineer for the last twenty years.

♦

F. P. ZIMMERLI, well known metallurgical engineer, has been selected to receive the Albert Sauveur Award given by the American Society for Metals "for his basic research in the field of shot peening to increase favorable stresses in the surfaces of metal parts".

♦

DR. C. H. MATHEWSON, Professor of Metallurgy at Yale University, has been elected to receive the Gold Medal awarded by the American Society for Metals for 1947 "for outstanding metallurgical knowledge and great versatility in the application of science to the metal industry, as well as exceptional ability in the diagnosis and solution of diversified metallurgical problems".

♦

H. C. CARROLL, formerly assistant engineer of the General Electric's Marine and Aeronautics Engineering Division, succeeds RAY STEARNS as engineer in charge of this division. Mr. Stearns has been associated with the company for forty-five years.

♦

ROBERT A. WEINHARDT, widely known automotive engineer, is now automotive power plant engineer of Willys-Overland. In his new capacity, Mr. Weinhardt, who during the war assisted in constructing the high-powered engines for the P.T. boat, will be in charge of all the company's engine design including a new 6-cylinder engine which is under way.

♦

WALTER L. STUTZ has retired as chief of the engineering instruments and mechanical appliances section of the National Bureau of Standards, Washington, after 35 years' service with the bureau.

♦

KARL P. HANSON, prior to becoming head of the mechanical engineering department at North Carolina State College, Raleigh, N. C., was professor of mechanical engineering, University of Connecticut.

ASSETS to a BOOKCASE

Elements of Mechanism

By Peter Schwamb and A. L. Merrill, late professors and W. H. James, professor emeritus, Massachusetts Institute of Technology; edition revised and rewritten by V. L. Doughtie, professor of mechanical engineering, University of Texas; published by John Wiley & Sons, Inc., New York; 428 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, $4.00 *postpaid.*

Possibly the oldest American text on mechanism — it was written in 1885—this book has a truly excellent balance of kinematics and mechanism analysis. The volume is a careful presentation in which not too much space is devoted to elementary concepts and the more complicated mechanisms are treated in detail.

This sixth edition brings the work up to date, while retaining the authors' approach to design. All chapters have been completely revised. A number of chapters including "Vectors," "Velocity Analysis" and "Acceleration Analysis," have been completely rewritten. New illustrative problems have been used throughout the text, and the discussion changed to conform with present-day practice and standards. In many cases theory has been developed and given practical applications to make the book more useful, and current terminology and examples have been used throughout.

The book has not been intended by the authors to serve as an encyclopedia of mechanisms. Rather, it was their purpose to select the relatively common and fundamental machine elements and to study their motions when combined in various ways. That they have achieved this purpose and met a great need is attested by the consistent use and demand for the book.

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The Chemistry and Technology of Resins

By Raymond Nauth, consultant in plastics; published by the Reinhold Publishing Corp., New York; 540 pages, 6 by 9¼ inches, clothbound; available through MACHINE DESIGN, $9.50 *postpaid.*

Outstanding book on plastics, this volume presents to the engineer detailed information on important aspects of most commercially available resins. Highly practical, it offers many tables of physical properties, numerous charts and diagrams and is replete with detail sketches and illustrations. Such salient aspects as proper selection of plastic materials for specific applications, molding problems, finishing problems and new processes of manufacture are discussed in detail. Information on chemical composition and manufacturing technique, while not occupying a disproportionate space, has not been neglected. Among the plastics on which new information is provided are synthetic rubbers and wood laminates. Types of materials covered include thermosetting plastics, amino resins, cast resins, amine-aldehyde resins, laminates, cellulose resins, synthetic rubbers and polyamides.

□ □ □

ASTM Standards, 1946

Published by the American Society for Testing Materials, Philadelphia; in five volumes, 7000 *pages, 6 by 9 inches, clothbound; available through* MACHINE DESIGN *postpaid; part II* $12.00, *other volumes* $8.00 *each, complete set* $44.00.

This 1946 edition of the American Society for Testing Materials' Standards is, because of the immense growth in the number and detail of standards, almost twice as large as the previous (1944) edition. Over 1400 specifications, tests, and definitions have been included. Published in five volumes, the standards are divided into three major sections. Part one, in two volumes, deals with ferrous metals (IA) and nonferrous metals (IB). Part two is a one-volume treatment of nonmetallic constructional materials. The remaining two volumes are concerned with other nonmetallic materials, the first, part IIIA, covering materials not commonly used in machine structures, such as coal, aromatic hydrocarbons, textiles and petroleum products. Part IIIB discusses the remaining nonmetallic materials, most of which are involved in design. They include electrical-insulating materials, plastics, rubber, paper and adhesives.

□ □ □

Analysis and Design of Translator Chains

By H. Ziebolz, vice president and chief engineer, Askania Regulator Co.; published by that company in Chicago; two volumes, 411 *pages,* 5½ *by* 8½ *inches, paperbound; available through* MACHINE DESIGN, $5.00 *postpaid.*

This discussion of a new approach to the design of mechanisms and machine controls reduce all designs to basic elements and establishes methods for solving complex problems. These basic elements for the conversion of force, energy or power have been called by the author "translators". It is shown that the variables involved in machine design, such as force or pressure, may in their entirety be as few as twenty-five;

How the First All-Steel Welded Vise is Made

By H. F. Seymour, President
Columbian Vise & Manufacturing Co.
Cleveland 4, Ohio

WHAT is believed to be the first all-steel vise fabricated by arc welding is the Columbian Work Shop Vise shown in Fig. 1. This vise was redesigned in collaboration with Parsons Engineering Corp., welding specialists, from conventional construction to arc welded steel to achieve superior strength, accuracy and long life.

Basic material used is SAE 1020 hot rolled steel plate, punched and formed to the proper shapes. Fig. 2 shows the component parts. The base plate is 3/8" thick,

Fig. 2. Component parts of the vise.

the nose piece 5/16" and all others 3/16".

The first fabricating operation is the welding of the front, or sliding, half. The rail, T-blank, front body and jaw are assembled in a jig and tack-welded. In

Fig. 3. Welding front body to rail with inside fillets.

this same jig the front body is welded to the rail with inside fillets (see Fig. 3), and the nose piece tacked on.

This front half is then passed to the

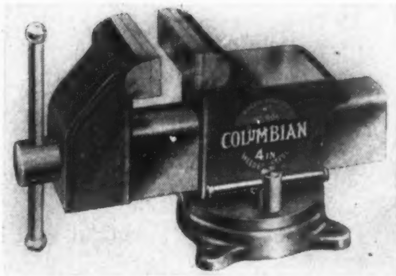


Fig. 1. The Columbian all-steel welded vise.

next welder, who makes the final fillets which join the nose piece and T-blank to the front body. The next operator welds the jaw to the nose piece.

Meanwhile, the rear half of the vise is being assembled in similar manner by another bank of welders. The rear body and swivel base are assembled in a jig and tacked together by the first man. The next man welds these parts with continuous fillets.

The front half and the partially-assembled rear half of the vise are then passed to a welder who fits them together. He inserts the sleeve, or inside guide, in the rear body, slides the rail of the front half through it, and welds the sleeve to the rear body with four 1-inch welds. Fig. 4 shows how the assembly is clamped in the jig. He then tack-welds on the rear anvil.

The vise then goes to the last welding operator, who welds the rear jaw to the anvil and the anvil to the rear body.

The vise assemblies are passed from one operator to the next by means of steel chutes.

Air hammers are used to peen the welds and clean off the spatter, and the welds are then ground smooth. Hardened steel jaw faces are drilled and counter sunk into the jaws. The screw and handle are forged steel and the malleable iron nut is replaceable.

AWS Type E-6030 electrode is used for welding the rear body to the base in

Fig. 4. Raising clamp which held inside sleeve for welding to rear body.

order to get a fillet with exceptionally deep penetration. For all other welds, E-6012 electrode is used.

With 12 set-up men and welders, 1,000 vises are produced a week. Production can be stepped up to 2,500 a week by increasing the force to 22 men.

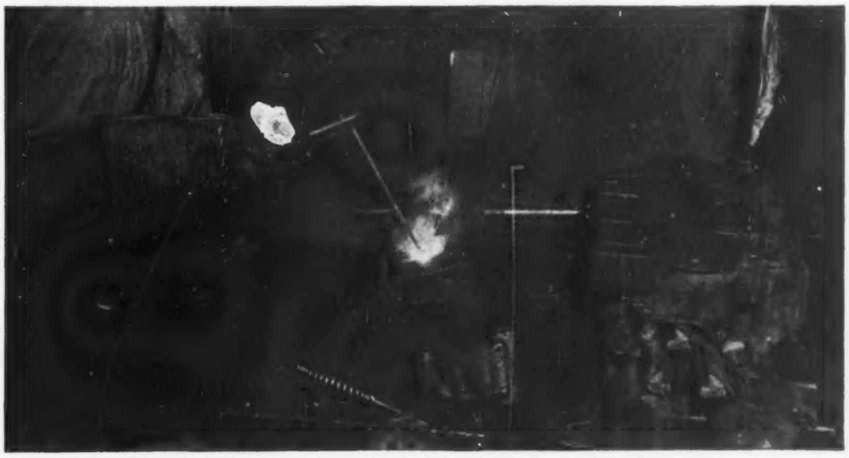


Fig. 5. Making final welds—anvil to body, and jaw to anvil.

the author proceeds to cross-tabulate them and identify each cross-point in the tabulation as a specific translator. He then shows how several of these translators may be used in series as a "chain" to achieve an ostensibly difficult result.

The profusely illustrated book, being an entirely new concept in mechanical design, is not easy to read. However, it promises much in the way of time saving and the understanding of mechanism to those willing to undertake its study.

□ □ □

Electronic Dictionary

By Nelson M. Cooke, executive officer, Naval Research Laboratory, and John Markus, associate editor, Electronics; published by McGraw-Hill Book Co., New York; 433 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$5.00 postpaid.*

Actually a compact encyclopedia, this volume should prove of value to engineers concerned in any way with electronics. Containing over 6500 terms and 600 diagrams, it covers the field from basic definitions to detailed descriptions of complicated apparatus. As an example, the abfarad is defined with respect to basic energy values as well as its relationship ($1/10^9$) to the farad. Descriptions and definitions are carefully made and clearly written and the volume is replete with illustrations.

Latest revised edition of the SAE handbook makes available war-developed data and new standards in addition to revised material carried over from the previous editions. Among new data featured are the first specifications for hydraulic brake fluid and standards for involute serrated shafts, filler and lubrication fittings and automotive steel castings. General information is also given on welding electrodes and copper and silver brazing. Priced at $5.00 for members and $10.00 for nonmembers, the 800-page clothbound volume is available from the Society of Automotive Engineers, 29 W. 39th St., New York 18.

Published under the title "Compressed Air Handbook" and representing the collective knowledge in the field of compressed air of the 19 member companies of the Compressed Air and Gas Institute, this handbook makes available new and original data not heretofore available in a single volume. Covering application, installation, operation and maintenance of compressed air systems, the book provides information on test standards, and includes many basic tables and formulas. The first seven chapters discuss representative installations, portable tools, installation design, and applications and the last chapter provides 114 pages of engineering data. Clothbound and measuring 6 by 9 inches, the handbook is available at $3.00 from the Compressed Air and Gas Institute, 90 West St., New York 6.

Report on German developments in the cold extrusion of steel has been published under the title, "Cold Shaping of Steel" by the Heintz Manufacturing Co. Prepared by the Heintz company as a report to the Army Ordnance Department, the 142-page, 8½ by 11-inch book describes a production process which has rarely, if ever, been used in this country. Yet, the process gives promise of radically changing our manufacturing thinking. Highly economical of material and production time, the technique may be used in the manufacture of many precision parts now made of malleable iron, gray iron, drop forgings or even parts now completely machined from bar stock —and do it without requiring any machining at all! Paperbound, the book is available at $3.25 from the Heintz Manufacturing Co., Front Street and Olney Ave., Philadelphia 20.

Adhesives

(*Concluded from Page* 126)

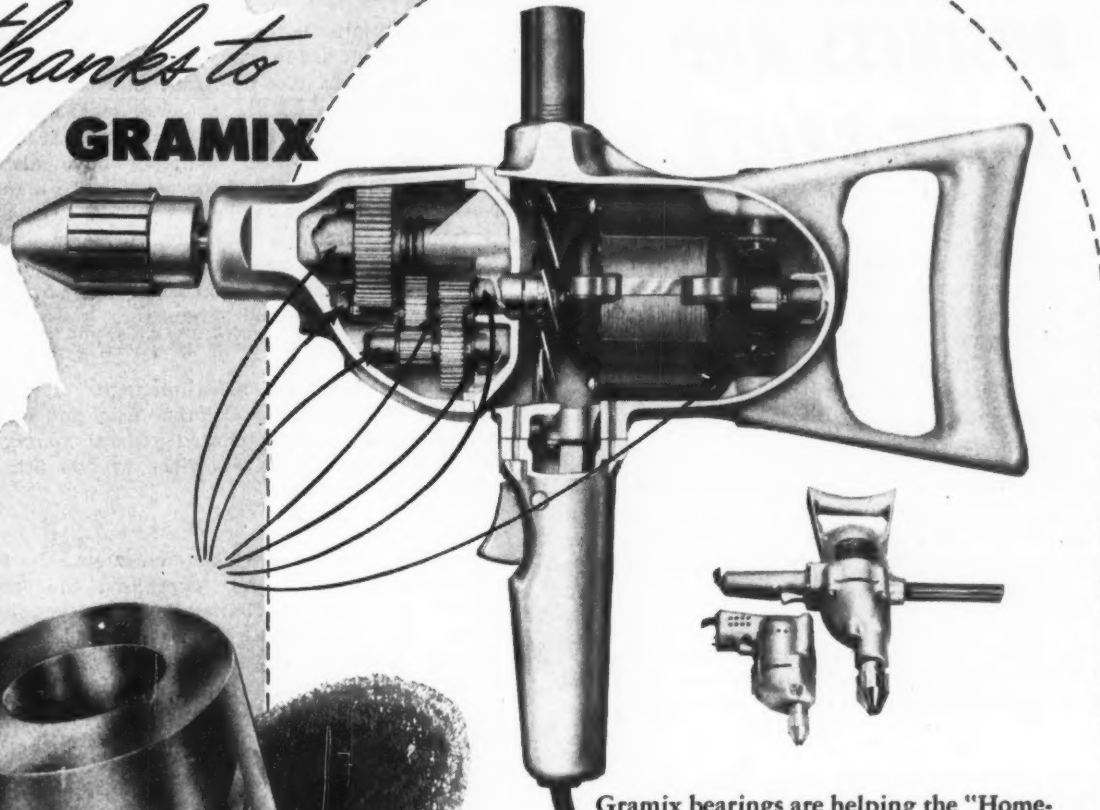
sheets which are effectively longer than existing presses and laminating equipment. It also permits the fabrication of many odd-shaped assemblies.

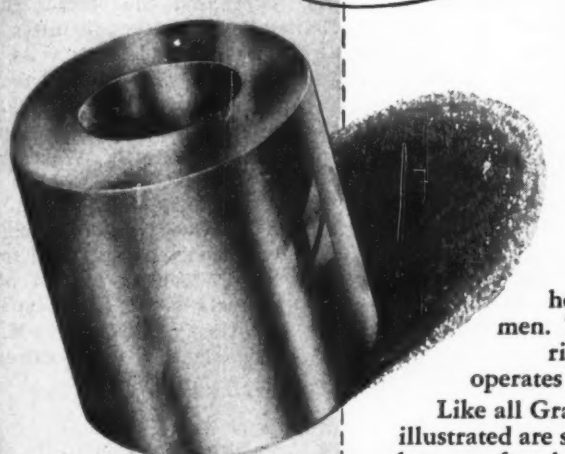
New resins of the type utilizing styrene or an allyl monomer with a polyester are becoming more useful in the making of insulating parts for electrical equipment. These are used largely as laminating resins in the preparation of solid structures from glass cloth, cotton cloth or paper. In such applications the resins must have good impregnating properties as well as good adhesive qualities. The making of a glass cloth laminate with such a resin involved the following procedure:

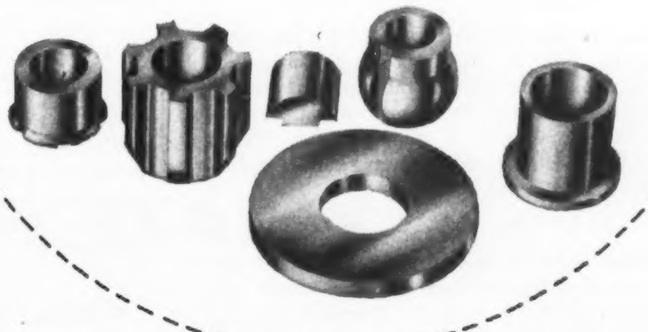
Liquid resin, properly catalyzed with a peroxide type catalyst, was brushed carefully onto sheets of glass cloth as they were piled up. It was important to apply the resin so that air bubbles were removed. After the piece was built up to the desired thickness, it was placed between heated press platens and a relatively low pressure of 5 to 25 psi was applied. The resin cured to form a hard structure.

Such compositions can be used for electrical insulation when temperatures run as high as 125 C continuously, and even higher intermittently. Joints can be prepared between such structures by using the same resin as for the laminating.

Valuable as they are at the present time, one of the most interesting things about these adhesives is the indication they give of future possibilities. Recent developments in the adhesive field are of major interest to the electrical industry, and it is important that chemists actively continue investigations to take full advantage of materials that are as yet little utilized but have wide potentialities.







BUSINESS AND SALES BRIEFS

NEW England office has been established in Hartford, Connecticut by the Ward Leonard Electric Co. District manager, located at 37 Webster St., is C. F. Shea, formerly attached to the main office at Mount Vernon, N. Y. In the Los Angeles area W. R. L'Hommedieu, with offices at 722 E. Washington Blvd., is the new sales representative.

Appointed manager of the newly established Allis-Chalmers branch office at York, Pa., G. Edward Conn Jr. will be assisted by MacGregor G. Jones. Mr. Conn has been a salesman in the Philadelphia district office since 1943.

Formerly located at 34 E. Logan St., Philadelphia, the Automatic Temperature Control Co. now has its offices and factory at 5212 Pulaski Ave., the same city.

Two new appointments have been made by Jack & Heintz Precision Industries. Albert A. Ricker, formerly vice president in charge of finance, has been appointed assistant to the president. He will at the same time retain the former office. Frank R. Kohnstamm, has been appointed general sales manager. Mr. Kohnstamm resigned duties as manager of the testing equipment division of Baldwin Locomotive Works to take up his new duties.

New laboratory testing service has been inaugurated by the American Chemical Paint Co., manufacturers of protective finishes. Service offered includes subjection of production samples of parts to rigorous tests at regular intervals. In addition, a weekly examination of data on the operation of cleaning and Alodizing baths is carefully studied and any irregularity reported to the manufacturer.

At the recent meeting of the Gray Iron Founders' Society in Milwaukee, H. A. Stockwell, of the Barbour-Stockwell Co., was re-elected president. Other officers elected were R. E. Kucher, vice-president, E. B. Smith, secretary, and H. J. Trenkamp, treasurer.

Two sales executives have been appointed by Westinghouse Electric Corp. John E. Payne, formerly manager of industrial sales, has been named manager of all industry sales departments for the corporation. R. S. Kersh, who has been manager of the Houston, Texas office since 1942, has been appointed manager of industrial sales to succeed Mr. Payne.

Licensing arrangements to manufacture and sell fittings have been made with two Los Angeles firms, The Pacific Screw Products Corp. and the Deutsch Co., by the Parker Appliance Co. Agreements concluded provide that the firms acquire rights to Parker's AN and 811 flared-tube couplings.

Two elections to the board of directors of the Air Reduction Sales Co. Inc. have been announced. Charles D'W. Gibson has been vice-president in charge of sales since 1937 and John A. Hill, formerly secretary of the company, has been a vice-president since 1945. Three appointments within the company have also been announced. Harold H. Reed has been made manager of the New York district succeeding W. S. Schoenthaler who is retiring. Oren M. Donohue, formerly assistant sales manager, has been appointed assistant manager, succeeding Mr. Reed, while William B. Brower now succeeds Mr. Donohue as assistant sales manager.

Formerly assistant sales manager of the Wickwire Spencer Division of the Colorado Fuel and Iron Co., H. C. Allington has been appointed general manager of sales. Mr. Allington will have offices at 500 5th Ave., New York City.

According to a recent announcement, the Syntron Co. of Homer City, Pa., has purchased the former H. K. Porter Co. plant at Blairsville, Pa. New facilities will enable the company to increase production in its line of shaft seals and other products. General offices will remain in Homer City. In Canada, an office has been opened in Montreal. Managed by C. F. A. Gray, the office will be located at 4695 Sherbrooke St., West.

Several new appointments have been made by Kennametal Inc. Jerome G. Brady is tool engineer and representative working out of the Pittsburgh district office and Lester Speickhoff will operate in the same capacity out of the Syracuse, N. Y. office. In Norway, K. W. Jonvik will represent the organization with offices at Dronningens gt. 17, Oslo.

With completion of a three-story addition to its quarters in Torrington, Conn., the Haydon Mfg. Co. will move all operations from nearby Forestville. The company will also cease activities in another two-story building in Torrington.

Industrial Sales and Engineering Co. have been appointed field engineers and sales representatives for the Ajax Flexible Coupling Co. Inc. Offices of the new representatives are at 352 Miners' Bank Building, Wilkes-Barre, Pa.

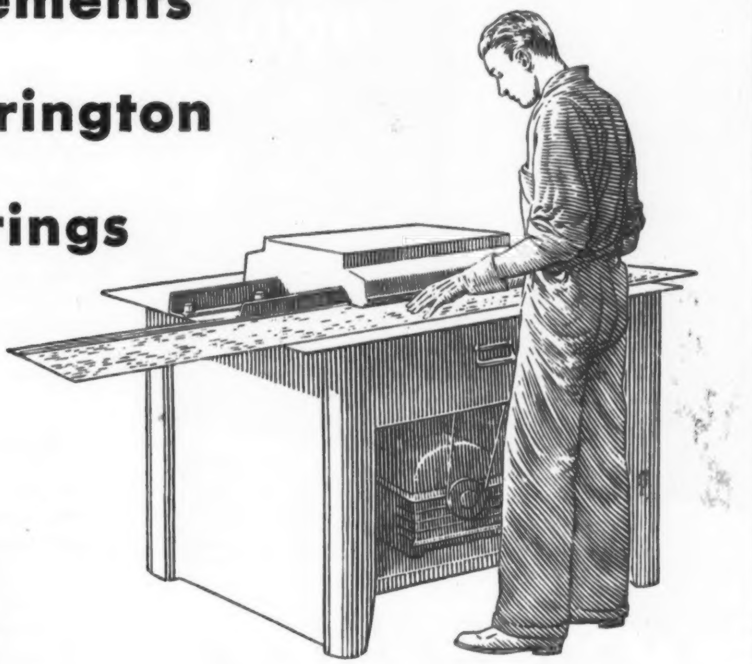
Previously manufacturers of specialized electrical equipment for use in automotive, marine and allied fields, the Leece-Neville Co., 5363 Hamilton Ave., N.E., Cleveland, has announced expansion into the electric-motor field. Present production is concerned with fractional-horsepower a-c and d-c motors.

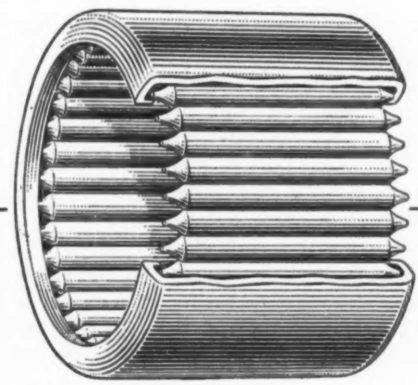
Formerly senior development man in the industrial products division of the B. F. Goodrich Co., Ernest Hookway has been appointed operating manager of the recently created materials sales division of that organization.

Following the recent death of C. W. Machon, general sales manager of the Brown & Sharpe Mfg. Co., P. R.

In one pass, Lockformer sheet metal machines form flanges ordinarily requiring many brake operations. Needle Bearings on gear trains and forming rolls greatly reduce power requirements and increase service life.

Power Requirements Cut 50% by Torrington Needle Bearings





... another advantage of these compact anti-friction units is capacity to carry high radial loads imposed on forming rolls of the Lockformer.

BY CHANGING to Torrington Needle Bearings on gear trains and forming rolls, the Lockformer Company cut the power consumption 50% in all models. *No major design changes were required* in adapting these compact anti-friction units to the limited space available. Simple housing design and easy installation by one arbor press operation keep fabrication and assembly costs low.

Ten years of successful operation in thousands of Lockformer machines attest the long service life assured by the high radial capacity, smooth operation and efficient lubrication of Needle Bearings.

These are only a few of the many advantages you can realize in equipment you design, build, sell or operate by insisting on Torrington Needle Bearings.

You will find the engineering skill and experience of Torrington's engineers helpful in adapting these efficient anti-friction bearings to your requirements. Call or write the nearest Torrington office or direct to...

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District Offices and Distributors in Principal Cities

TORRINGTON BEARINGS

NEEDLE
SPHERICAL ROLLER
STRAIGHT ROLLER
TAPERED ROLLER
BALL
LOOSE ROLLERS

Hatch has been appointed sales director. Associated with Mr. Hatch will be H. L. Merrill, W. H. Spence and W. E. Anderson.

Appointment of O. T. Clarke as dealer representative in the southeast territory has been made by the Hammel-Dahl Co. of Providence, R. I. Mr. Clarke will make his office in Decatur, Ga.

With offices in Louisville, Harry E. Weiler has been appointed manager of the Reynolds Metals Co. sales office for that district, serving all of Kentucky except Kenton and Campbell counties. In addition, southern Indiana is served by this office.

Facilities of the Superior Steel Corp. will be expanded shortly by the installation of a new 3-high roughing mill which is to be used in conjunction with the modernization of their hot-mill facilities at Carnegie, Pa. New mill will enable the company to increase available widths of clad metals.

Election of a new vice president has been announced by Acme Aluminum Alloys, Inc. of Dayton, Ohio. In the capacity of executive vice-president, B. D. Claffey will assume the duties of general manager of the corporation.

Charles Bruning Co. Inc. has opened a branch office in Cleveland. The new Bruning office is located at 213 W. St. Clair Ave., and is under the direction of E. M. Elstad who formerly represented the company in northern Ohio.

New plant has been constructed by the Bodine Electric Co. for the manufacturing of its type U split-phase motor. Mass production of the new motor is expected to make available better deliveries on large-quantity orders.

Appointment of David G. Henderson as sales engineer has been made by the Hannifin Corp. Mr. Henderson will cover the complete Hannifin line of hydraulic and pneumatic equipment in the Pittsburgh area. He can be contacted at P.O. Box 8002.

According to a recent announcement, William E. Bryden, sales engineer, has been transferred by the Timken Steel and Tube Division of the Timken Roller Bearing Co. from their Chicago office to their Cincinnati office. He will be succeeded at Chicago by William T. Strickland.

Three changes have been made in the sales organization of Fairbanks, Morse & Co. J. C. Elmburg, manager of the company's Boston branch, has been transferred to the Atlanta, Georgia office to assume the position of manager of that area. V. O. Harkness, previously manager of the diesel sales division at Chicago, has been appointed manager of the Boston branch, and T. M. Robie of Chicago has been appointed to the position of manager of the general diesel sales division.

Formerly with the Republic Steel Corp., Frederick C. Shotwell has joined the Cold Metal Product Company's Detroit sales force. His offices will be in the Curtis Building. Samuel M. Marshall, also of the sales division,

Kodak

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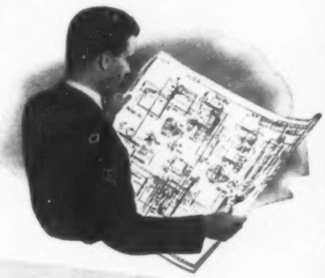

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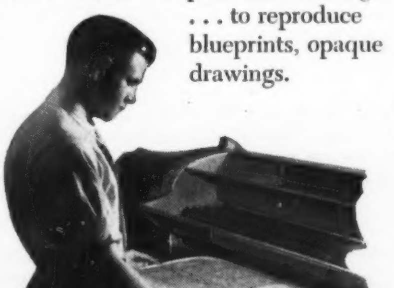

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has been transferred to the New England offices with headquarters in Hartford, Conn.

♦

Appointment of Robert C. Graves as vice-president in charge of sales has been announced by Federal Electric Products Co. Mr. Graves was previously vice-president in charge of sales of the Trumbull Electric Mfg. Co.

♦

Jones & Laughlin Steel Corp. has announced a plant improvement program entailing the investment of more than $100,000,000. The program covers all departments of the corporation and completion is scheduled by the end of 1949.

♦

With headquarters at 1659 W. Market St., Akron, the Industrial Products Sales Co., has been appointed Ohio representative of the Standard Tube Co. Operating from this office will be H. William Kranz Jr. and Edmund Siess Jr.

♦

Two promotions have been announced by Tube Turns, Inc. Norton P. Bosemer has been placed in charge of the Los Angeles office and Robert S. Tyler Jr. is now in charge of the newly established mid-continent district. Headquarters of this new area are in Tulsa at 311 Tuloma Building.

♦

Name of Aircraft Screw Products Co. Inc. of Long Island City, New York, has been changed to Heli-Coil Corp. The address is unchanged.

MEETINGS AND EXPOSITIONS

Jan. 26-29—

Institute of Aeronautical Sciences. Sixteenth annual meeting to be held at Hotel Astor, New York. Robert R. Dexter, 2 East 64th St., New York 21, is secretary.

Jan. 26-29—

Refrigeration Equipment Manufacturers Association. Fifth all-industry refrigeration and air conditioning exposition to be held at Public Auditorium, Cleveland. F. J. Hood, Clark Bldg., Pittsburgh, is chairman of the show.

Jan. 26-30—

American Institute of Electrical Engineers. Winter general meeting to be held at William Penn Hotel, Pittsburgh. H. H. Henline, 33 West 39th St., New York 18, is secretary.

Feb. 2-6—

American Society of Heating and Ventilating Engineers. Fifty-fourth annual meeting and eighth international heating and ventilating exposition to be held at Grand Central Palace, New York. Charles F. Roth, Grand Central Palace, New York 17, is manager of the exposition.

March 3-5—

Society of Automotive Engineers Inc. National passenger car and production meeting to be held at Hotel Book-Cadillac, Detroit. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

March 15-19—

American Society of Tool Engineers. Sixteenth annual meeting and sixth annual industrial exposition to be held at Cleveland Public Auditorium, Cleveland. Harry E. Conrad, 1666 Penobscot Bldg., Detroit 26, is executive secretary.

Welded Trailbuilder

(*Concluded from Page* 148)

welding stations are combined with machining stations. The first operation consists of cutting the cylinders to length from long lengths of tubing. Next is welding the manifolds to the cylinder. The cylinder then is taken to a specially constructed boring machine and bored to within 0.005-inch of final size. After boring, the cylinder is taken to a special lathe which is tooled up to machine threads at the forward end of the cylinder. These threads are used to hold the front cylinder head. From there the cylinder is taken to a three-spindle drill press where the manifolds are drilled and tapped. The cylinder is then put into a fixture and taken to a vertical honing machine which hones the inside of the cylinder to within plus or minus 0.0005-inch.

Following the honing operation the bottom head is pressed into the cylinder. This bottom head is made up of a machine torch-cut block jig welded to a round machine torch-cut plate 1-inch thick. This subassembly is machined to press fit into the bore of the cylinder. Fixture for the machining of this head locates on the trunnion block and a J-groove is turned in the head for better weld penetration. A hole is then drilled in the trunnion block which is the trunnion of the cylinder.

After this head assembly has been pressed into the cylinder it is located in an automatic submerged-arc welding machine and welded. This welding machine welds the heads on to the cylinders in about forty seconds. Because of this welding procedure and rate of welding, it has been possible to eliminate all distortion due to welding. After welding, the cylinder does not change shape more than plus or minus 0.001-inch and, therefore, no additional machining operations are required because of welding.

The top cylinder head is an iron casting which screws into the cylinder. In this new design the casting is much smaller and less expensive to machine than the one used with the cast cylinder. This new style cylinder also allowed realization of shop savings in the machining of the piston, piston rod and piston rod nut. Clevis of the cylinder also is fabricated. It is made up of a piece of machine torch-cut bar stock drilled and tapped for connection between the piston rod and the top ears of the lift arm. Finally, the assembled cylinders are put in a hydraulic testing machine and tested up to 1500 psi for leaks. By combining machining operations with welding operations on the same production line, it was possible to speed up manufacturing of these cylinders considerably, with a resultant reduced manufacturing cost.

With the new welded unit the contractor has a machine which is lighter in weight, and the resultant maneuverability and better control make it possible for him to move more yards of dirt per hour. This decrease in weight also allowed him to carry less dead weight on the tractor and reflected itself in the payload. Carrying less dead weight on the tractor also cuts down his cost of fuel, lubrication, etc.

NEW MACHINES

And the Companies Behind Them

Air Conditioning

Fan. For equalizing temperature and humidity, and for drying and cooling. Particularly adaptable in cold storage, milk cooling and meat storage rooms. Available in two sizes: AA 20-in. delivering 3000 cfm and AA 24-in. delivering 4800 cfm. Chelsea Fan & Blower Co. Inc., Irvington 11, N. J.

Automotive

Ozone machine. For garages, weighs 50 lb, operates on ordinary 110-volt alternating current, and purifies air in areas up to 6000 sq ft within 40 min through the production of ozone. Perfection Stove Co., Cleveland.

Ceramic

Tunnel kiln. For small batch production work, firing of ceramic parts, and heat treating of small parts on a production basis. Heating element provides maximum temperature of 2200 F. Kiln is of pusher type, with small refractory trays pushed through inside of 33-in. firing chamber by a variable speed drive. K. H. Huppert Co., Chicago 37.

Domestic

Radio-phonograph combination. Has fast, jamproof automatic record changer playing twelve 10-in. or ten 12-in. records. Equipped with 10-in. electrodynamic speaker. Crosley Division, Avco Mfg. Corp., Cincinnati.

Radios with ac-dc chassis. Four models available, all featuring a newly-designed speaker, full-view slide-rule dial, and full broadcast range, 540 to 1600 kilocycles. Crosley Division, Avco Mfg. Corp., Cincinnati.

Refrigerator. Combination refrigerator and home freezer, unit has two separately refrigerated, completely insulated compartments, each with its own door and control. Upper compartment maintains temperature of zero F for freezing foods and storing frozen foods; lower compartment maintains about 38 F for normal fresh food storage. General Electric Co., Bridgeport 2, Conn.

Radio-phonograph combination. The unit incorporates 12-tube FM-AM circuit, 12-in. permanent magnet speaker, 4 complete bands, and record changer. Crosley Division, Avco Mfg. Corp., Cincinnati.

Built-in television units. Designed to blend with rooms, offices, clubs, hotels, lounges, etc. Custom built, the set has a screen almost as large as a full-size newspaper page. RCA Victor Division, Radio Corp. of America, Camden, N. J.

Wringer-washer. Equipped with built-in timer. Porcelain enameled, or aluminum self-draining tub holds 8 lb of dry clothes. General Electric Co., Bridgeport 2, Conn.

Television set. Featuring "Swing-a-view" picture tube and incorporating AM-FM radio chassis and automatic record changer. Crosley Division, Avco Mfg. Corp., Cincinnati.

Gas furnace. Incorporates principle of radiant heat within the firebox. Iron Fireman Mfg. Co., Portland, Oreg.

Dairy

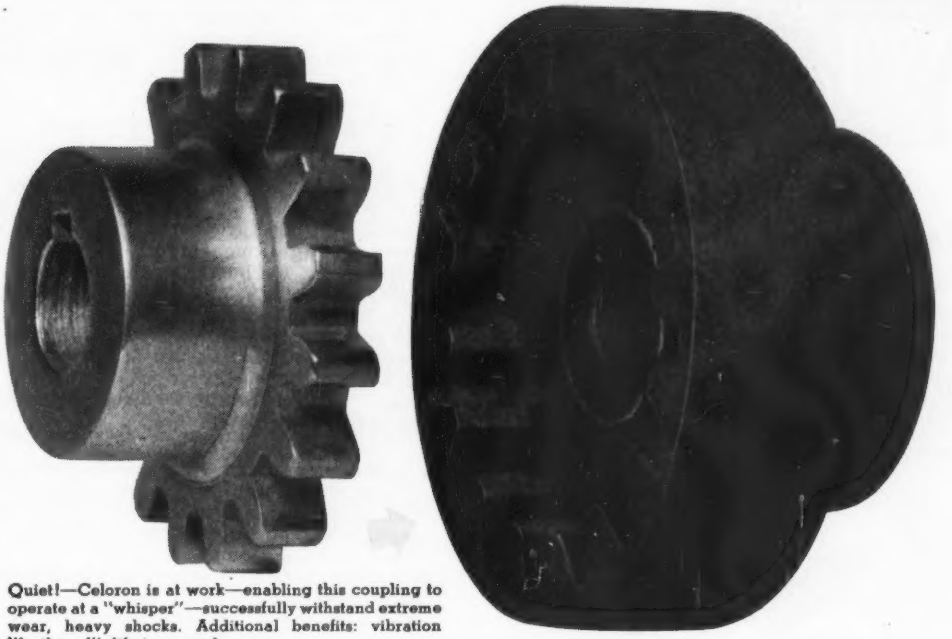
Automatic washer. For milking machines. Operating on vacuum supplied by milker vacuum pump, it rapidly draws a cleansing solution through the teatcup assemblies. Then automatically reverses the direction of solution flow through the milker parts. Advance Engineering Co., Waukesha, Wis.

Heating and drainage system. For steam-heated equipment. 8-in. x 48-in. accumulator will handle drainage from as many as twelve machines, or units with total steam consumption to 30 hp. Fred H. Schaub Engineering Co., Chicago 23.

Separator. Simple attachments convert single-bowl unit into a milk clarifier or standardizer. Three models available. Rite-Way Products Co., Chicago 13.

Finishing

Gyra-Flow finishing machine. For washing, rinsing, drying, dipping, spraying and baking in one continuous



Quiet!—Celoron is at work—enabling this coupling to operate at a "whisper"—successfully withstand extreme wear, heavy shocks. Additional benefits: vibration "feathered", life increased.

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cycle, in standard sizes having from two to twenty interchangeable operations. George Koch Sons Inc., Evansville 4, Ind.

Spray gun. Production type, for all types of finishes. Spray pattern is adjustable from round to flat with all intermediate patterns. Binks Mfg. Co., Chicago 12.

Metallizing gun. Nozzle incorporates new development of mixing gas and oxygen at the point of combustion. Features fingertip turbine control. Vande Mfg. Co., Chicago.

Heat Treating

Laboratory furnace. Temperatures up to 3000 F, and for continuous operation as high as 2500 F. May be used as a muffle furnace, or with carbon blocks for nonoxidizing atmospheres. Lindberg Engineering Co., Laboratory Furnace Division, Chicago 12.

Induction heater. Coil designed to function with any size bearing race less than 8⅜ in. OD, accommodating all sizes up to 7-in. journal diameter. Hyatt Bearings Div., General Motors Corp., Harrison, N. J.

Salt bath furnace. For cyclic annealing alloy steel forgings. Entire process requires 10 to 60 min. The Ajax Electric Co., Philadelphia.

Industrial

Heat transfer unit. For use in connection with X-ray equipment. Dissipates heat generated by X-ray tube. Capacity is rated up to 1000 watts, depending on operating temperature range. Eastern Industries Inc., New Haven 6, Conn.

Dispensing tanks. Indirectly heated and thermostatically controlled. Built on double-boiler principle, the two new types feature a series of oil-carrying tubes running through the length and width of inner vat. Aeroil Products Co. Inc., West New York, N. J.

Chemical cleaner. Portable, electrically-heated unit for on-the-spot indoor cleaning of industrial machinery, conveyor systems, engines and other equipment. Hartman Corp. of America, St. Louis.

Tank-type cleaner. For general vacuum cleaning, dust collecting and water pick-up. Incoming air is whirled around inside tank and heavy particles are deposited there; air passes through cloth bag filter where dust is removed; and collected dirt is emptied by lifting out the 9 gal tank and dust bag. Ideal Industries Inc., Sycamore, Ill.

Instruments

Electron diffraction instrument. For studying surfaces and thin films of metals, ceramics and plastics. Special Products Div., General Electric Co., Schenectady 5, N. Y.

Laboratory

Laboratory mill. For grinding oil bearing seed and nut samples and industrial products. Operates at 3600 rpm, with 3 or 5 hp motors, or arranged for V-belt drives. The Bauer Bros. Co., Springfield, O.

Laundry

Presses. For shirt finishing with flush pushbutton controls, adjustable for fingertip operation. Prosperity Co. Inc., Syracuse 1, N. Y.

Materials Handling

Fork truck. For handling loads of four to eight thousand pounds. Features controller which functions as electric counterpart of automotive automatic gearshift, high-pressure hydraulic system, and silicone insulation and lubrication. Automatic Transportation Co., Chicago 20.

Crane truck. For construction, maintenance and handling. Has full 40-ft boom, constructed in 10-ft sections. Center sections are removable, providing 20-ft and 30-ft booms where required. Yale & Towne Mfg. Co., Philadelphia.

High-lift truck. 3000 lb capacity, with four-wheel suspension, auto steer and towing eye. Lyon-Raymond Corp., Greene, N. Y.

Booster. Portable belt conveyor type. Is capable of being fed or discharged from either end. Island Equipment Corp., New York 17.

Power-belt conveyor. Standardized, multipurpose type for moving medium weight loads between floors or horizontally. The Rapids-Standard Co. Inc., Grand Rapids 2, Mich.

Metalworking

Single-spindle automatic turret machine, with electric tool feed drive making variable forward and return tool feeds possible without cam changes for each of five

positions in turret head. Cleveland Automatic Machine Co., Cincinnati.

High-pressure hydraulic die casting machine. Has small over-all dimensions and is capable of locking dies with high pressure to produce castings with a minimum of flash. Cleveland Automatic Machine Co., Cincinnati.

Power index type milling machine for machining exhaust and intake manifolds. Capable of producing 90 pieces net, per hour. Has three stations. The Cross Co., Detroit.

Cylinder block main bearing straddle facing machine. Carries feed out of roughing cut and back again for finishing. The Cross Co., Detroit 7.

Presses of 150-ton capacity, with a bed size of 41 x 84 in., weighing 56,000 lb. Also available in models of 100, 200, 250 and 300-ton capacity in a variety of bed sizes. Die Tool Engineering Co., Detroit 3.

Tapping machine with lubricating pump of motor-driven gear type providing continuous flow of lubricant to tap. Procunier Safety Chuck Co., Chicago 6.

Lathes with quick-change mechanism providing fingertip selection of fifty-four threads and feeds, forty-five of which are obtained by shifting two levers on the gear box and the remaining by changing position of a sliding gear. Atlas Press Co., Kalamazoo.

Mining

Dual mud screen (shale shaker). Screen 36 x 48 in., driven individually by V-belts from a single electric motor, steam turbine, gas or diesel engine. Link-Belt Co., Chicago.

Refining

Sediment removal and transfer conveyor for cleaning settling tanks. Conveyor is basically an endless link-chain, motor-driven, on which 4-in. rubber flights are mounted at 6 in. intervals. Honan-Crane Corp., Lebanon, Ind.

Rubber

Duster-deduster for automobile inner tubes. Injects into tube at the extrusion die a carefully proportioned stream of compressed air and soapstone. Whittington Pump & Engineering Co., Indianapolis 4.

Portable belt measuring and cutting machine. Handles belts up to 30 in. wide and 2000 lb in weight. Operates in space of 12 x 12 ft. Ralphs-Pugh Co., San Francisco 5.

Disperser and homogenizer for pigments and other solids in all vehicles including rubber and latex. Tri-Homo Corp., Salem, Mass.

High-speed dissolver and disperser. One model has capacity of 100 to 500 gallons. Cowles Co., Cayuga, N. Y.

Printing

Web offset jobber designed to operate at speeds from 10,000 to 30,000 lithographic impressions an hour with a fixed sheet cut-off of 8½ in. R. Hoe & Co. Inc., New York.

Testing

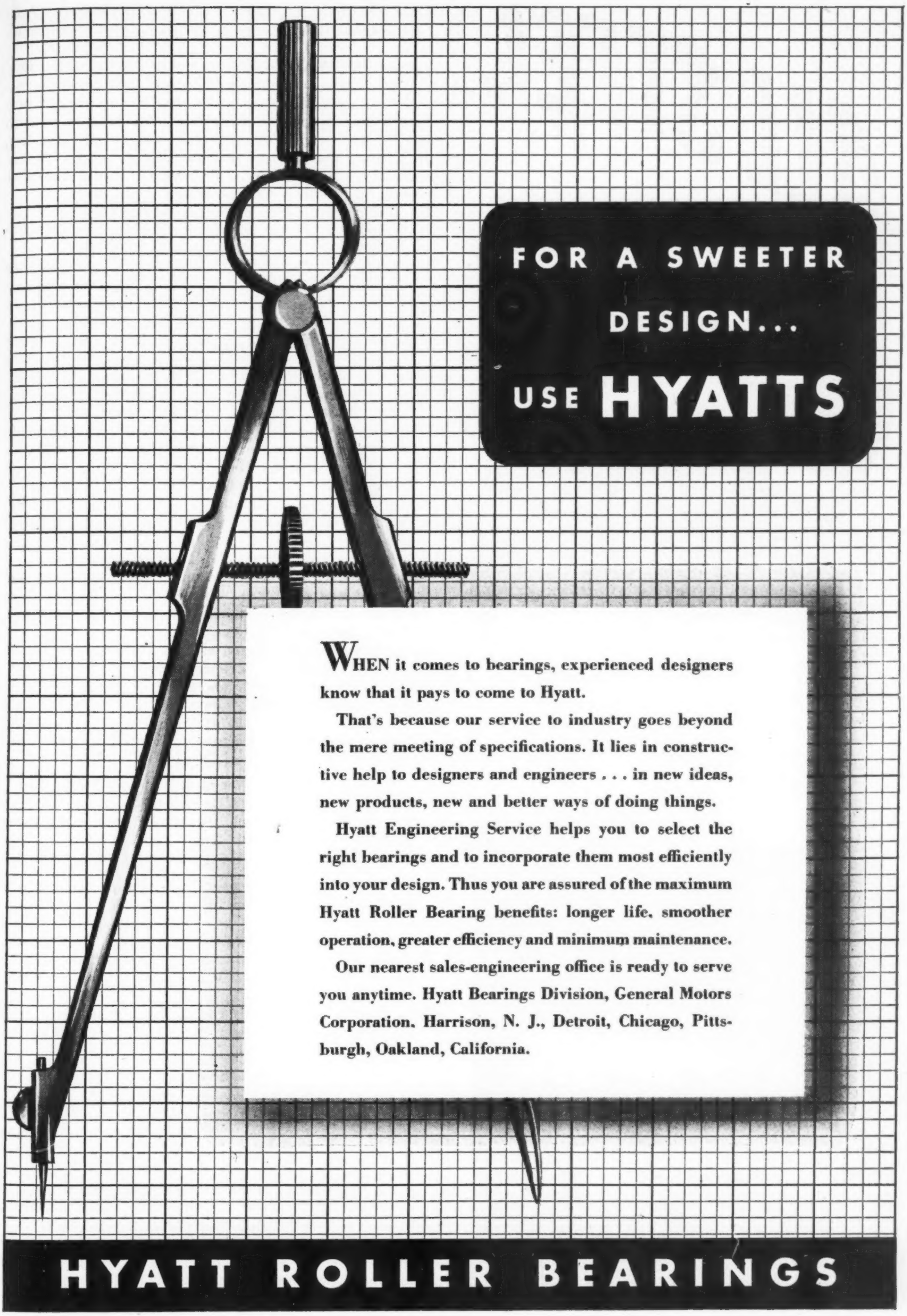
X-ray photometer for determining the tetraethyl lead content of gasoline, the concentration of an acid in water, the per cent chlorination of a plastic, or the per cent ash in coal. General Electric Co., Schenectady 5.

Physical tester for handling long specimens, as well as specimens that show considerable stretch. Accomplished by lengthening uprights to give a 13-in. spread between grips. W. C. Dillon & Co. Inc., Chicago 44.

Motor-driven stiffness tester designed for testing light metals, foil, paper, plastics, textiles, leather, hard rubber, and other thin flexible sheet materials. W. & L. E. Gurley, Troy, N. Y.

Textile

Combination machine for scouring, washing and dyeing without removing material. Pneumatic controlled. Rodney Hunt Machine Co., Orange, Mass.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HYATT ROLLER BEARINGS

New type spinning frame for making spun rayon from continuous filament rayon. Medley Mfg. Co., Columbus, Ga.

Four-section, full-fashioned hosiery knitting machine. Features a semi-automatic welt-turner, full automatic friction controls, individual motor drive and up to eight yarn carriers. Gages available are 45, 51, 54, and 60 with round or square heels. Robert Reiner Inc., Weehawken,

Tumble jar tester, gear reduction motor driven. Dimensions: 14 x 22 x 15 in. Motor operates on 115 volts ac or dc. Andrew Technical Service, Chicago.

Machine for detwisting, detacking, opening and squeezing. Uses electric eye to detect right or left twist. Cook-P & N Machine Co., Boston 27.

Quick-drying moisture determination unit for plant or laboratory use. Unit will receive an 8-in. diameter drying pan of 1, 2 or 3-in. depth. Temperature ranges from 150 to 300 F. Harry W. Dietert Co., Detroit 4.

Machine for continuous dyeing, slashing and beaming of cotton yarns. 2700 ends of 9s yarn can be processed through machine at rate of 25 to 30 yards per minute, with similar speeds for other yarn and warp size. United Steel of America Inc., New York.

Motor-driven label gummer for labels of any length up to 8½ in. in width. Has adjustable screw for determining amount of glue. Driven by a 1/20-hp motor. Diagraph-Bradley Stencil Machine Corp., St. Louis 8.

Automatic bobbin refinishing machine for small lot refinishing operations where problems arise from rough and nonuniform hand-refinished bobbins. Refinishes 10 bobbins per minute. J. M. Nash Co., Milwaukee.

Vacuum cleaner which uses water to trap material picked up; designed for textile mills since lint neither clogs any of the mechanism nor decreases vacuum power. Rexair Inc., Rockville Centre, N. Y.

Fork truck for lifting cotton bales and similar bulky loads to a height of 17 ft. Towmotor Corp., Cleveland 10.

Single or double machine for wrapping rubber, plastics, wire or other yarn-like materials. Drive is from an integrally mounted electric motor through V-belt. Spindle speeds are adjustable from 1000 to 1500 rpm. Canister Co., Phillipsburg, N. J.

Vending

Automatic fruit vendor, refrigerated. Will deliver correct change and a cold apple, orange or pear. Kold Krisp Apple Service Inc., Van Nuys, Calif.

Wire

Rod and wire pointer. Consist of a pair of grooved gap rolls and a cut-off shear driven through a combination gear and V-belt drive by a standard constant-speed ac motor. Aetna Standard Engineering Co., Youngstown,

Continuous cure wire insulating extruding machine, with a 2¼-in. bore. G. & V Machine Co. Inc., Paterson, N. J.

Wire coating machine for radio tubes. Six-furnace type with individual pyrometer control. Eisler Engineering Co. Inc. Newark 3, N. J.

Woodworking

High-speed medium duty router. Two models are available. One single-speed, 20,000-rpm machine and the double-speed 10,000 and 20,000 rpm. Onsrud Machine Works Inc., Chicago 47.

Electric edge gluer has automatic cycling control. Pneumatic feed permits operation at greater speeds. Integral mounting of diametric heater reduces radiation interference possibility. Bell Machine Co., Oshkosh, Wis.

Continuous edge gluing machine. Produces selected joint pressure up to 740 psi on 4/4 stock, the full width of machine. Feed variable from 2½ to 7½ fpm. James L. Taylor Mfg. Co., Poughkeepsie, N. Y.

DESIGN ABSTRACTS

Design Simplification Pays

ONE desgner in a hundred has a thorough knowledge of production methods. The other ninety-nine would be embarrassed if asked to explain why the fabricating processes they call for in drawings are most economical. Small wonder that hordes of requests for drawing changes "to facilitate production" pour into the engineering department even while production starts.

But most important cost consideration is not the material nor production methods. "Is it necessary?" is what the designer should ask himself when considering a new part. Obvious way to cut cost of a part is to eliminate the part itself. It also saves the expense of attaching it to adjacent parts. This overall saving must be matched against a possible cost increase due to manufacturing complexity of the remaining parts.

Costs Reduced

Not only do these preachings sound well, they work. Tooling and manufacturing costs can be reduced at no sacrifice in that all-inclusive factor called quality. Results from practicing this policy in design of a current model in production at Beech are proof of its efficacy. This airplane's designers maintain they've improved, not reduced quality. In many instances weight has been saved, none added. Light-gage structures have been replaced with heavier gages, less susceptible to vibration and fatigue.

In just ten items, 65 hr 50.34 min per airplane were saved in manufacturing time. Mulitplying total per plane savings by a yearly production quantity of 2000 airplanes shows a saving of 131,678 man-hours.

Experience on this job showed that the many years of complex structural design have instilled great resistance to change in most individuals. But once the simplification concept makes inroads in an engineering department, it becomes contagious.

Today the aircraft industry is in the position of the prisoner of long ago. After many years behind the bars, he one day leaned on the door to his cell and found it unlocked. It had always been unlocked. The way to freedom had always been his. And the way to improved aircraft at lower cost has always been the industry's.—*From a paper by W. A.*

Davidson, P. H. Pelley, R. E. Saunders and J. W. Rix, Beech Aircraft Corp., presented at the recent SAE National Personal Aircraft meeting in Wichita, Kans.

Planned Predesign Research

DURING the last fifty years a rapid evolution in our industrial economy has taken place, resulting from our rapidly expanding technical know-how. The point of view of the buying public has gone through a corresponding evolution as a result of a greater understanding of the basic sciences, a wide experience with manufactured products, and the educational forces of modern advertising. The evolution has brought into existence a discriminating buyer who has established prejudices, definite likes, and a real sense of values.

These and many obscure factors of market taste and market needs are vital controls over the product of a manufacturing establishment. They must be the guides and signposts by which both management and the engineer direct their product-development work. They are so vital to the success of this work that they can no longer be established by supposition or guesswork as in the past. No manufacturer today can afford to disregard these signposts and assume that by intensive advertising, product prestige, or merchandising trickery he can change them. Money invested in such an attempt is largely wasted and brings little or no return to the stockholder.

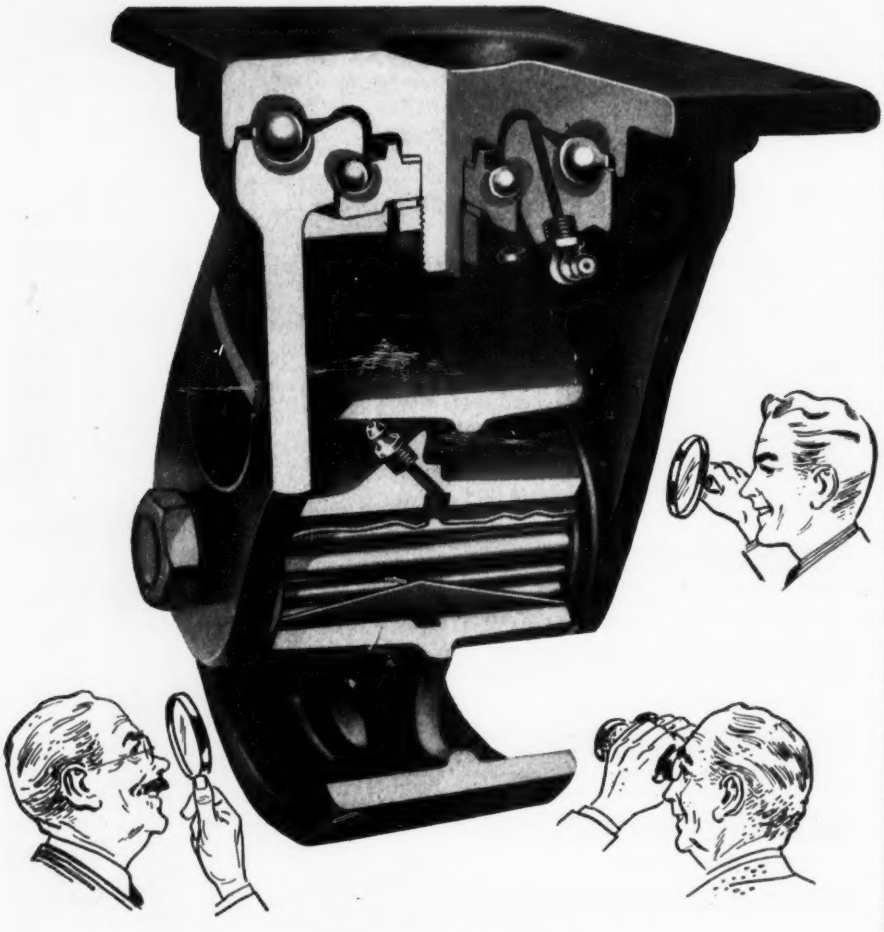
Three Steps Outlined

The problem, therefore, becomes specifically—how can a manufacturer, or a product engineer, ascertain the character and taste of the market for which he is designing.

Three steps, in our experience, are essential in the collection of accurate facts upon which the final statement of the problem can be based:

1. Field product research, which means a study of the problem; evolution of a pattern of questions which will secure the desired information; and a tabulation of this information with interpretation of the pertinent facts revealed.

2. An independent analysis made by qualified people with a completely objective attitude. In this analysis certain statistical facts from the field study may be utilized. However, this work should not be bound by past practice, or practical considerations of cost, or manufacturing limitations. The object of this work is to arrive



at ideal requirements regardless of whether or not they can be practically interpreted immediately.

3. Correlation of the facts revealed from the field study with the creative development ideas evolved from the analysis. This correlation results in ideal specifications and recommendations which act as bench marks to guide development engineering and to integrate the requirements of the final design of the product, thereby eliminating much waste of money in undirected development work.

Following these three primary steps, the ideal specifications must be modified from the point of view of practical manufacturing requirements, requirements of trade standards and trade practices, limits of manufacturing cost, and finally in terms of expected trends within the industry.

When this work has been prepared completely for presentation to the interested parties, it is possible to replace individual opinion with actual provable facts before any engineering work or development work is undertaken.

It has been our experience that by using such a logical system of arriving objectively at the facts and the complete statement of the problem, the work of the development engineer or designer is greatly simplified. The results he can produce are far superior to anything he can accomplish without it.—*From a paper by Roger L. Nowland, Van Doren, Nowland, and Schladermundt, presented at the recent ASME Annual Meeting in Atlantic City.*

User Research Discloses Unsuspected Attitudes

RESULTS of user research often disclose attitudes of potential customers which are of considerable importance in deciding on specific features of the product under consideration. For example, a manufacturer supplying a certain type of processing equipment to the chemical industry planned to redesign his product completely and in accordance with the preferences and requirements of its prospective buyers. Before any field investigation, he was particularly interested in what kind of driving mechanism they wanted and why, and what they thought about various possible means of power transmission. Informal interviews with engineers and production and maintenance personnel in chemical plants disclosed that they cared very little about specific com-

ponents of the unit, but stated their requirements in terms of application—flexibility of speed, economy as against other types of processing, adequate power, durability, resistance to shock load, compactness, and a minimum of maintenance. Interpretation of these general application requirements into specific terms indicated the need, for example, of a wide range of speed variation, such as had not been available in commercial makes of this equipment.

Another manufacturer was considering the manufacture of a certain type of communicating system which possessed features not included in other types available on the market. Some units of the new product—hand-made jobs—had been installed a few years ago, and there had been no complaints from the users. When the field investigator called on these users without disclosing the nature of his interest, he learned that the specific device which appeared to constitute the most important feature of the product was shut off, and for years had not been used because the operating personnel found it too complicated. There had been no dissatisfaction with the product—they just had turned one part off quietly and forgotten it.

Appraising the Market

Besides being instrumental in helping a manufacturer to design a new product "for use," user research may disclose facts and conditions which have a definite bearing on the number of units of a new product which a manufacturer can hope to sell. In a specific case, the product in question was a new type of supplementary equipment to be utilized by the textile printing and engraving trade. The product had unquestionable merit from an engineering point of view, and appeared to be a likely means of saving to the user. It was discovered, however, not only that the number of establishments which could use the device was small, but also that their operating capital did not permit them to make the necessary investment even if it presented a money-making proposition. Thus, an accurate appraisal of the potential market for the product prevented the manufacturer from undertaking the development and fabrication of a new product for which there were not enough buyers who could pay the price.—*From a paper by Ernst E. Wachsmuth, James O. Peck Co., presented at the recent ASME Annual Meeting in Atlantic City.*